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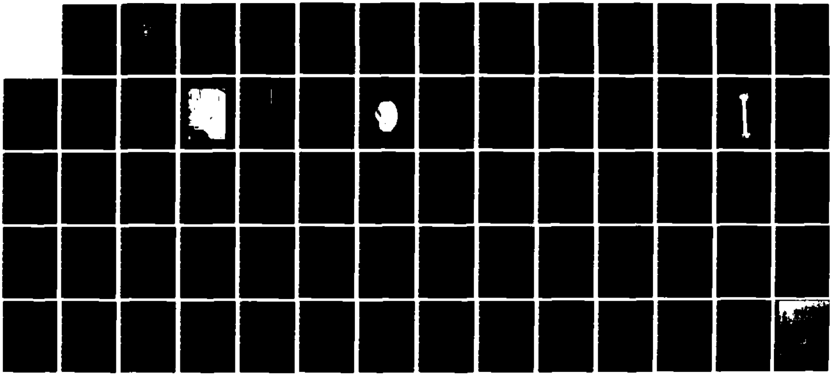

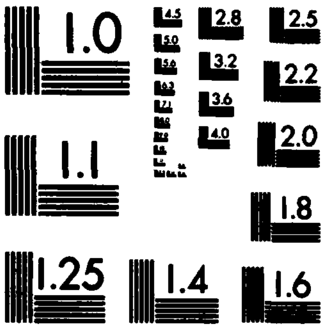

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THESIS

AN EXPERIMENTAL EVALUATION OF THE
THERMAL PERFORMANCE OF A ROTATING HEAT PIPE
WITH INTERNAL AXIAL FINS

by

George H. Gardner III

June 1983

Thesis Advisor: P.J Marto

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The finned condenser heat transfer rates were 30-40 percent greater than those of the smooth condenser. All data appeared to be influenced by the presence of noncondensable gases. Recommendations for improvements to the rotating heat pipe system are included.

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An Experimental Evaluation of the
Thermal Performance of a Rotating Heat Pipe
with Internal Axial Fins

by

George H. Gardner III
Commander, United States Navy
B.S., United States Naval Academy, 1968

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

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June 1983

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ABSTRACT

A rotating heat pipe was tested using two different copper condensers (a smooth cylinder and a cylinder with 22 straight axial fins) and water as the working fluid. The smooth condenser was tested at rotational speeds of 700, 1400 and 2800 RPM. The finned condenser was tested at 700 RPM. The heat transfer rate for each run was measured and plotted against the temperature difference between the internal vapor and the cooling water inlet. Representative condenser wall temperature profiles were plotted for each run. The primary objective was to compare the heat transfer rates obtained.

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I. INTRODUCTION

A. THE ROTATING HEAT PIPE

The rotating heat pipe is a closed device that can be used to transfer thermal energy radially or axially in a rotating machinery component. The basic configuration of the heat pipe used in this experiment is shown in Figure 1.1. It consists of a cylindrical evaporator, a cylindrical condenser and a small amount of volatile working fluid used to transfer energy from the evaporator to the condenser.

During operation, the heat pipe is rotated about its axis with sufficient speed to maintain a fluid annulus in the evaporator. Heat conducted through the evaporator wall vaporizes a portion of the working fluid, creating pressure and density gradients which cause the vapor to flow into the condenser section. The gradients are maintained by the condensation of the vapor on the condenser wall. Heat is removed by a second working fluid that cools the outside surface of the condenser. Condensate flow back to the evaporator is induced by a hydrostatic pressure gradient resulting from the axial difference in the condensate film thickness along the condenser wall.

B. BACKGROUND

The original rotating heat pipe concept developed by Gray [Ref. 1] in 1969 was a single circular cylinder. The rotating heat pipe at the Naval Postgraduate School (NPS) was designed by Daly [Ref. 2] in 1970. it had a stainless steel evaporator and a stainless steel truncated cone (3 degree half angle) condenser. Subsequent work has changed the evaporator to copper and various condenser geometries

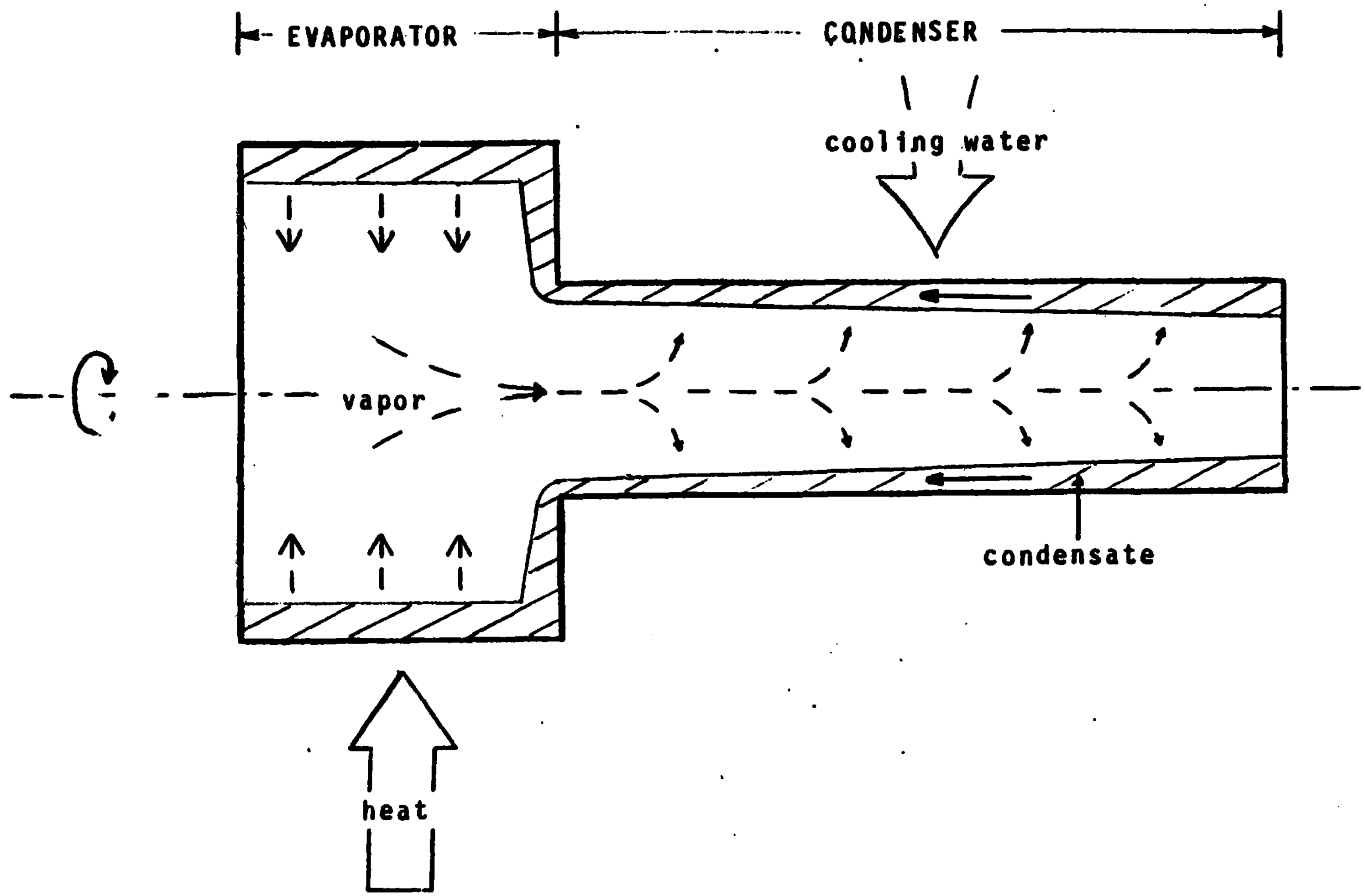

Figure 1.1 The Rotating Heat Pipe

have been tested at NPS with various working fluids. The best heat transfer rates have been achieved with a truncated cone condenser or a cylindrical condenser with the interior surface enhanced with helical fins [Ref. 3] and [Ref. 4]. For industrial application, the cost of the helical fin enhancement is less expensive than the cost of manufacturing the truncated cone condenser. Of all the working fluids tested in the previous work, water exhibited the best heat tansfer characteristics over the broadest range of operating conditions [Ref. 3]. Throughout the previous work, it was evident that the heat transfer at the interior surface of the condenser is the limiting factor in improving the thermal performance of the rotating heat pipe.

C. PRELIMINARY OBJECTIVES

The NPS heat pipe had been idle for three years prior to the commencement of this work. The system was therefore disassembled and inspected. As a result of the inspection, the following preliminary objectives were accomplished to prepare the system for operation.

1. Programs for computer aided data aquistion and analysis were written and tested. This was required since the data aquisition system used in this thesis had not been used with the rotating heat pipe in the past.

2. The temperature sensing system was changed from Copper-Constantan (type-T) to Chromel-Constantan (type-E) thermocouples. This was done to improve the sensitivity on the temperature measurements.

3. The cooling water drain system could not be located so a new one had to be designed and manufactured.

4. Other minor modifications to individual components are discussed in chapter II.

D. THESIS OBJECTIVE

The primary objective of this thesis was twofold. First, to reproduce the experimental results of Weigel [Ref. 3] and Wagenseil [Ref. 4] using a smooth copper condenser and water as the working fluid. This was done so that new results from this work could validly be compared to the results of previous work. Second, to experimentally determine the heat transfer characteristics of a condenser made from a commercially available copper tube having 22 evenly spaced straight axial fins on the inside surface.

The straight fin geometry was selected for evaluation because it had not been tested in the past. The results of the straight fin condenser could provide a baseline against which future work could be compared. The future work might include evaluations of condenser geometries with different size fins, different numbers of fins, different helix angles for the fins or different condenser sizes. The final goal being to determine the optimum rotating heat pipe condenser geometry.

II. EXPERIMENTAL EQUIPMENT

The rotating heat pipe system and ancillary equipment used in this thesis are shown in Figures 2.1, 2.2 and 2.3. The entire heat pipe assembly is bolted to a steel bed-plate that can be oriented from a horizontal to a vertical position.

A. EVAPORATOR

The evaporator (Figure 2.1) is a copper cylinder 100 mm in diameter and 70 mm long. It is sealed at both ends by O-rings.

B. HEATER

The heater (Figure 2.1) is a Chromel-A wire with Magnesium oxide insulation in an Inconel sheath. The heater is wrapped around the evaporator and thermally insulated on the outside. Electrical power to the rotating heater is supplied through four pairs of carbon brushes riding on a pair of bronze collector rings. The heater resistance is 1.8 ohms measured at the collector rings.

C. POWER SUPPLY

The power supplied to the heater is regulated by a voltage sensing circuit. The single phase, 440 volt, 60 hertz line voltage is fed into a precision differential voltage attenuator which divides the line voltage by one hundred. This stepped down voltage passes through a true root mean square (TRMS) converter stage on which the integration period is 1ms. The output of the TRMS converter is

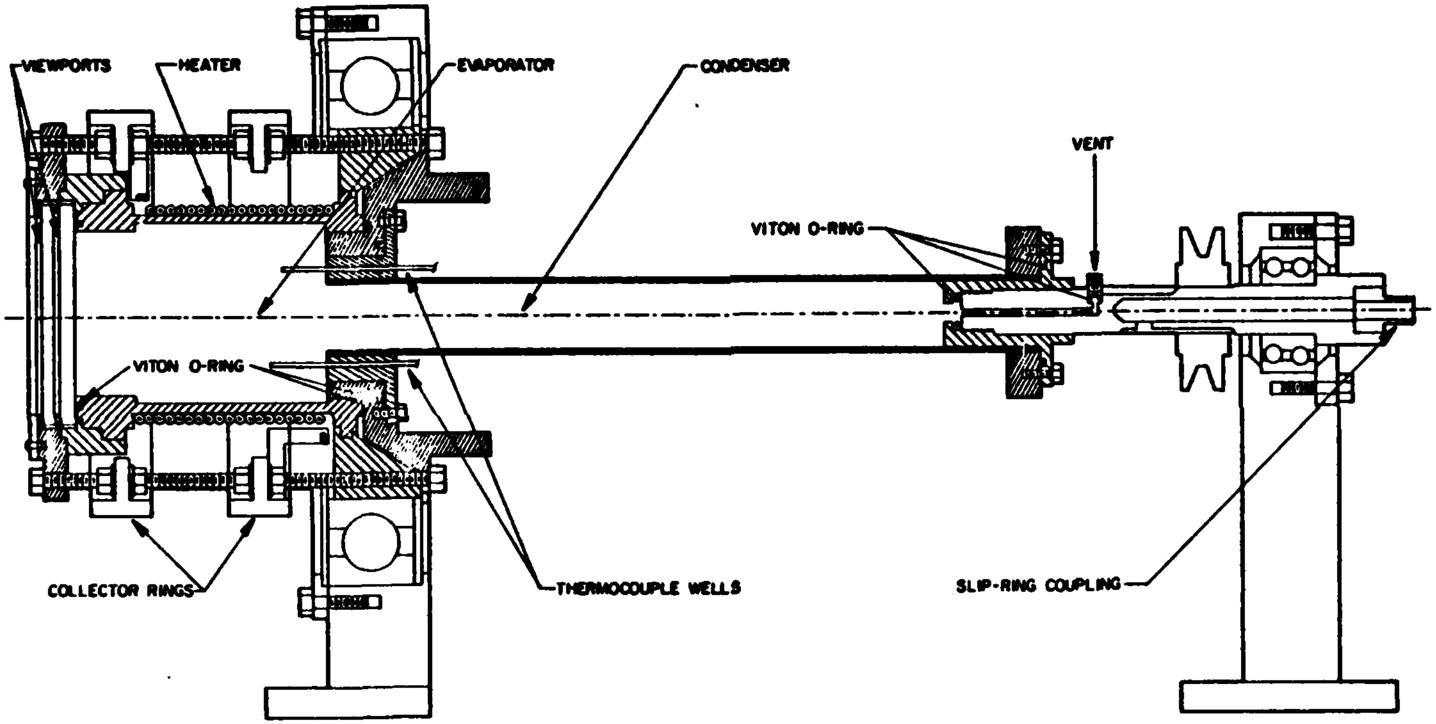

Figure 2.1 Details of the Rotating Heat Pipe

buffed and compared to the Power Control potentiometer mounted on the control panel. The comparator output is fed to the control input of a Holmar Silicon Rectifier which supplies the amplified voltage to the collector rings.

The output of the TRMS converter is amplified and filtered to provide a voltage proportional to the actual voltage supplied to the heater. This voltage is monitored by the voltmeter on the data aquisition system.

D. VIEW WINDOWS

Two 88.9 mm diameter 6.35 mm thick Pyrex glass windows are used to seal one end of the evaporator and allow observation of the heat pipe interior during operation.

E. BEARINGS

The rotating heat pipe system is supported by two bearings. The main bearing is provided with external cooling water coils and is lubricated by a small oil dripper (Figure 2.2) which is adjusted to provide an oil flow of 2-3 drops per minute. The drive pulley bearing is a self-lubricated sealed bearing.

In all previous works, the drive pulley bearing and the slip ring support (Figure 2.3) were separately mounted on the bed-plate and were connected by a flexible coupling. This arrangement required disassembly, reassembly and realignment each time a condenser was changed. For this work, the drive pulley bearing support and the viscous slip rings were aligned and mounted on a common plate that is then bolted to the bed plate. This new arrangement simplifies the assembly, and prevents undue bending or tension stresses from being placed on the delicate slip ring wiring that passes through the flexible coupling.

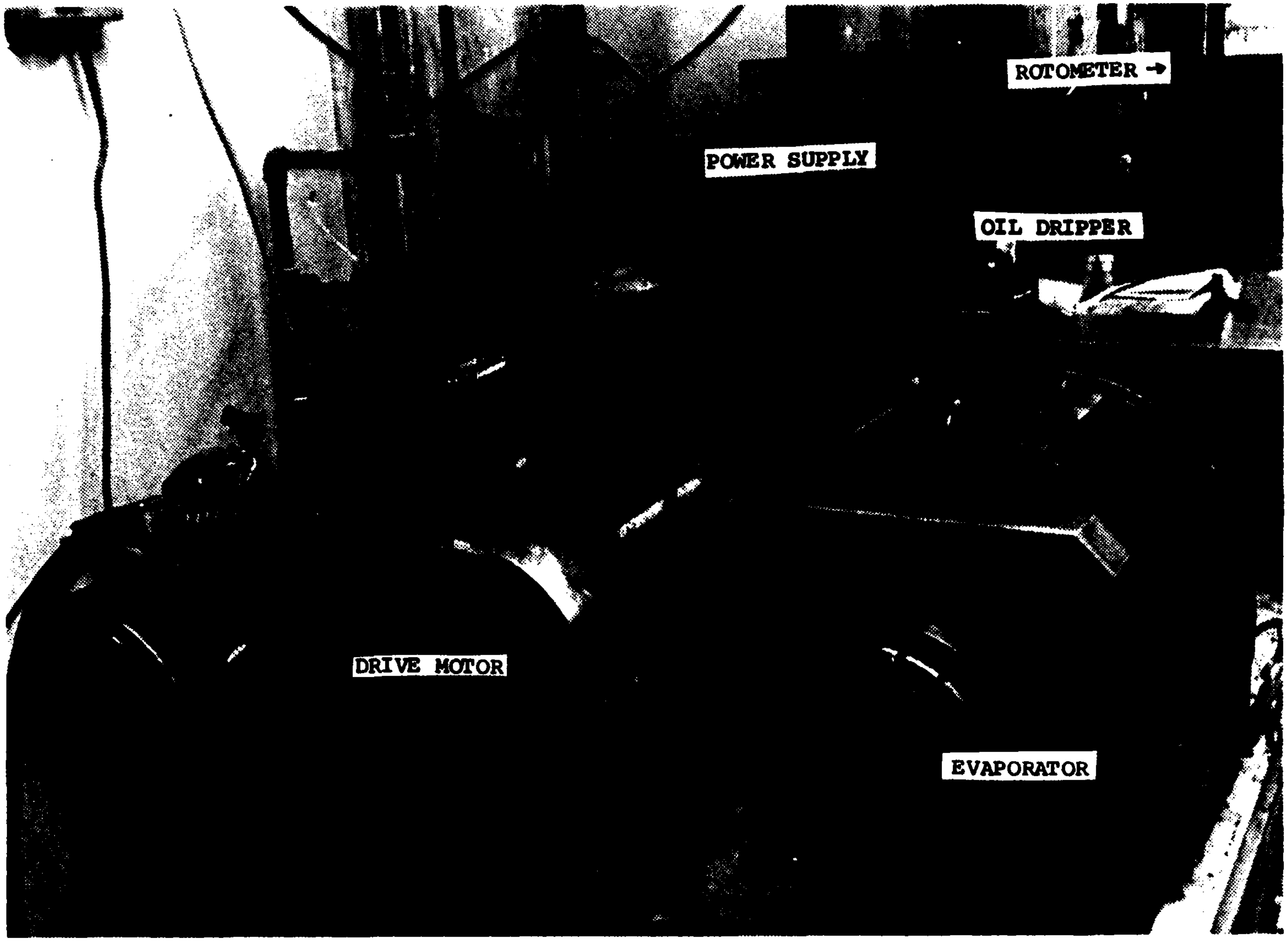

Figure 2.2 Photograph of the Rotating Heat Pipe System

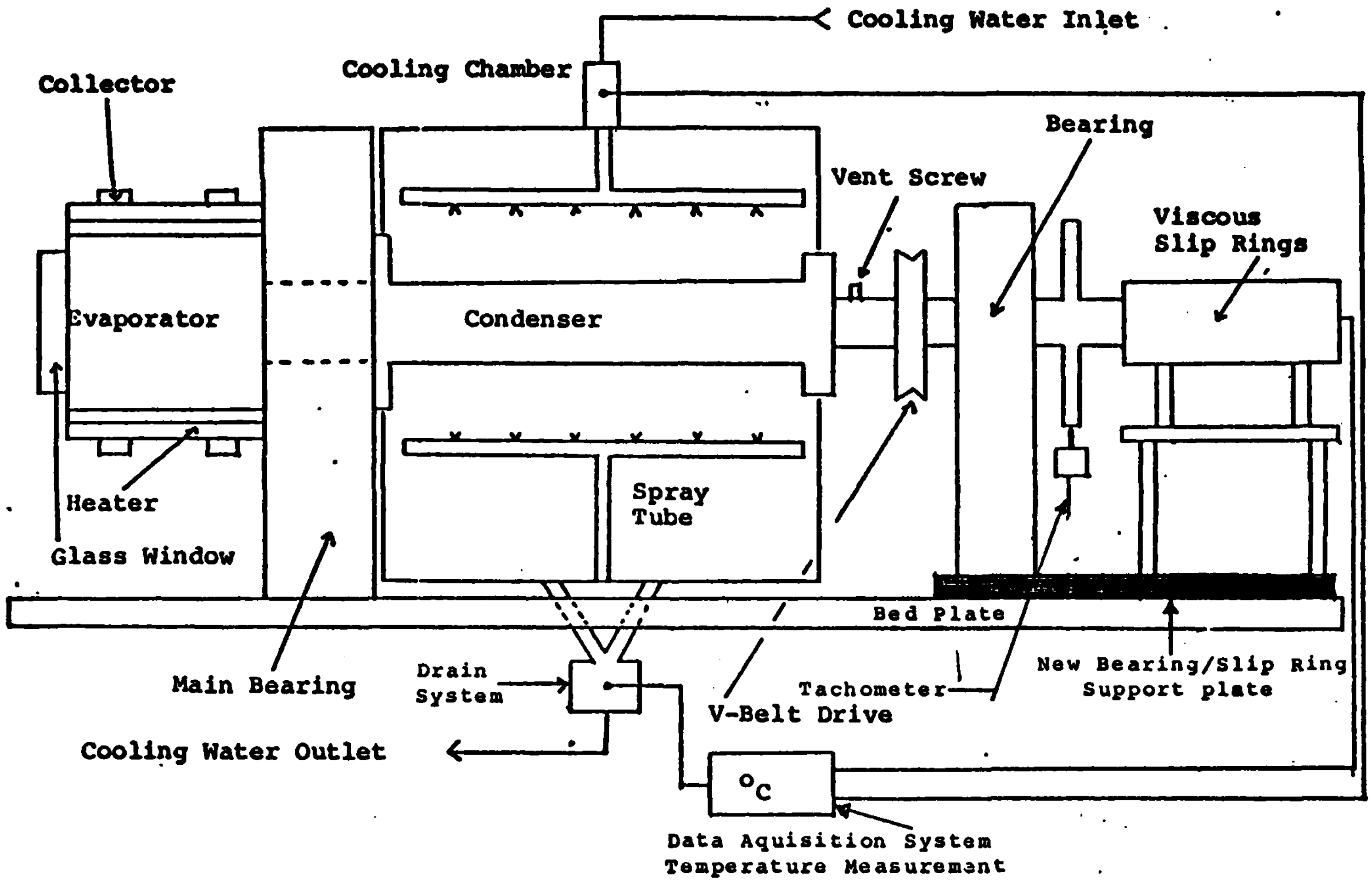

Figure 2.3 Experimental Set-Up

F. CONDENSERS

For this work, the condenser construction technique developed by Wagenseil [Ref. 4] was used with minor modifications. Two holes were drilled in the evaporator end flange and 1.6 mm diameter stainless steel thermocouple sheaths were inserted and silver soldered in place. The sealed end of the thermocouple sheaths extend into the evaporator for monitoring the vapor temperature in the heat pipe. With this construction, each condenser has two permanantly installed vapor space thermocouples whose wiring is not subjected to the bending and breaking that occurred when the vapor space thermocouples were installed in the main bearing flange.

Two copper test condensers were assembled for this thesis. Each is manufactured from a copper tube 295 mm long. Spray cooling takes place over a section 250 mm long.

1. A smooth cylinder the same as used by Weigel [Ref. 3] and Wagenseil [Ref. 4].

   Outer diameter: 26.9 mm
   Wall thickness: 0.75 mm

2. A smooth cylinder with 22 axial fins evenly spaced on the inside surface. The tubing, Figure 2.4, was provided by the Noranda Metal Industries of Newtown, Connecticut.

   Outer diameter: 28.6 mm
   Wall thickness: 1.3 mm
   Number of fins: 22
   Fin height: 1.4 mm
   Fin thickness: 1.4 mm

Area ratios (compared to the smooth condenser)

outer: 1.06
inner (negelecting fins): 1.02
inner (fins included): 1.80

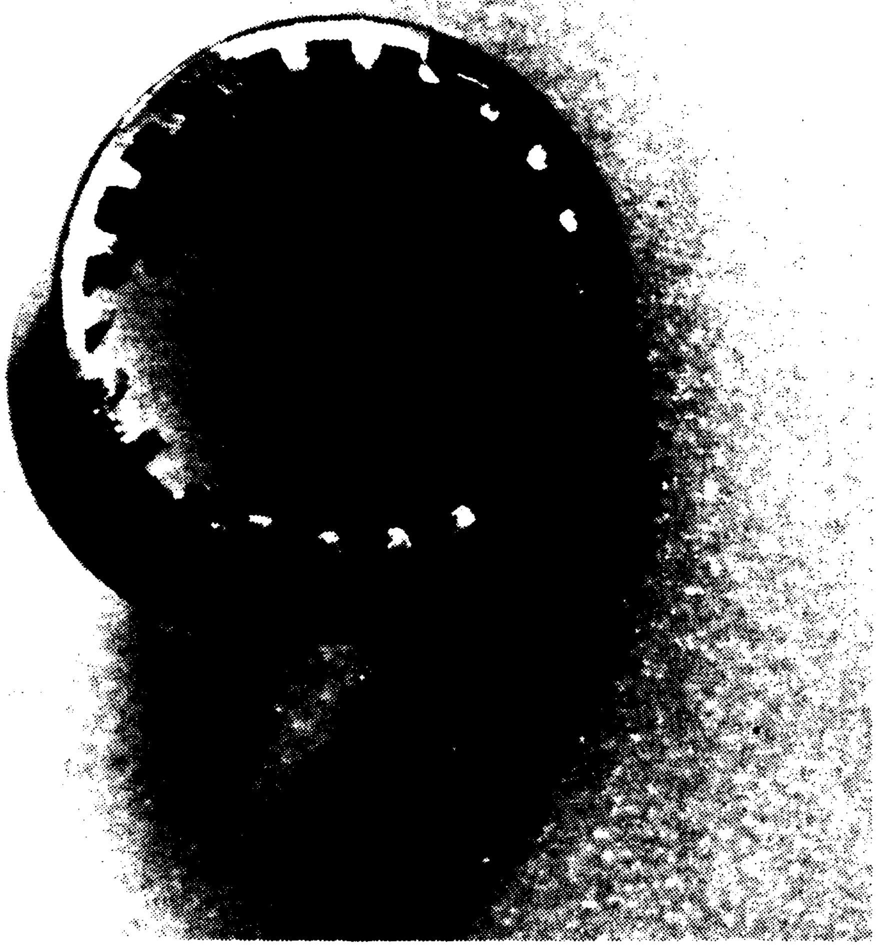

Figure 2.4 Finned Condenser Tubing

G. COOLING SYSTEM

The condenser is spray cooled by filtered and softened tap water. The cooling water is sprayed uniformily along the entire length of the condenser from four perforated tubes placed at 90 degree intervals around the condenser axis. The cooling water drips off the rotating condensers and is collected by two drains that feed into the new drain collecting box. The drain collecting box was designed to provide for good mixing in the drain path so that the water temperature measured at the collecting box outlet would be a true bulk temperature. The cooling water flow rate is measured by a calibrated rotometer.

H. HEAT PIPE DRIVE SYSTEM

The rotating heat pipe is driven by a v-belt attached to a variable speed electric motor. The RPM is measured by an induction tachometer and provides a digital display. Speed can be controlled within +/- 1 RPM.

I. VACUUM AND PRESSURE TEST SYSTEM

The vacuum test system shown in Figure 2.5 is used to determine whether or not the heat pipe is vacuum tight. The test flange is installed in place of the view windows and the manometer is used to monitor the vacuum when the system has been evacuated and isolated. For pressure testing the heat pipe, the compressed gas pressure regulator is attached to the test flange in place of the vacuum test system.

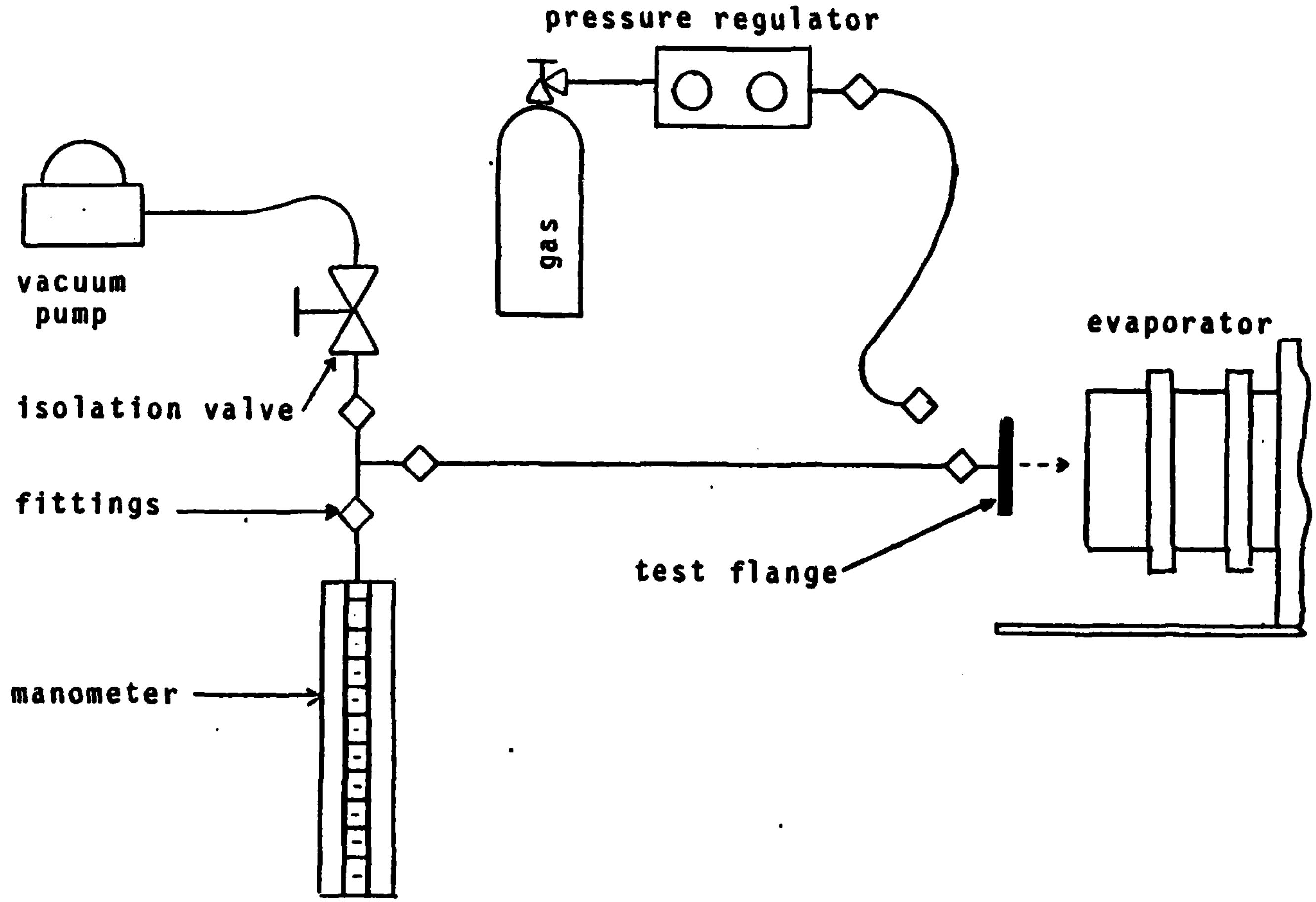

Figure 2.5 Vacuum and Pressure Test System

J. INSTRUMENTATION

The instrumentation used in this thesis is described in the following paragraphs. The stated accuracy for each instrument is the result of the calibration procedures described in Appendix B.

All temperatures were measured by welded type-E thermocouples made from 0.13 mm diameter wire with teflon insulation. A nylon sheath was placed over the wires to protect them from bending and abrasion.

The condenser wall thermocouples were soft soldered into grooves machined into the outer surface of the condenser. Figure 2.6 shows the axial position of the thermocouples. The accuracy of thermocouple placement is +/- 3mm axially and +/- 1mm radially. The uncertainty in the placement is caused by the easy bending of the thin wires, difficulties encountered getting solder to adhere to the type-E wire and problems in keeping the thermocouple bead fully immersed in the molten solder pool at the time of attachment.

The vapor space thermocouples were inserted into the wells until they came in contact with the end of the well. All thermocouple wires were held to the condenser at regular intervals using wire wraps. The thermocouple wires were passed through holes in the flange at the drive end of the condenser and were soft soldered to an EIA 25 pin connector. Spurious voltages were not expected in this connector. Since all the junctions were exposed to ambient air, the speed of rotation kept the connector cooled to ambient so that no thermal gradients occurred during operation. The poor soldering characteristics of the type-E wire resulted in numerous cold solder joints on this connector. The combination of cold solder joints and centrifugal force caused many thermocouple wires to lose electrical contact or pull out of this connector during operation. Repairing

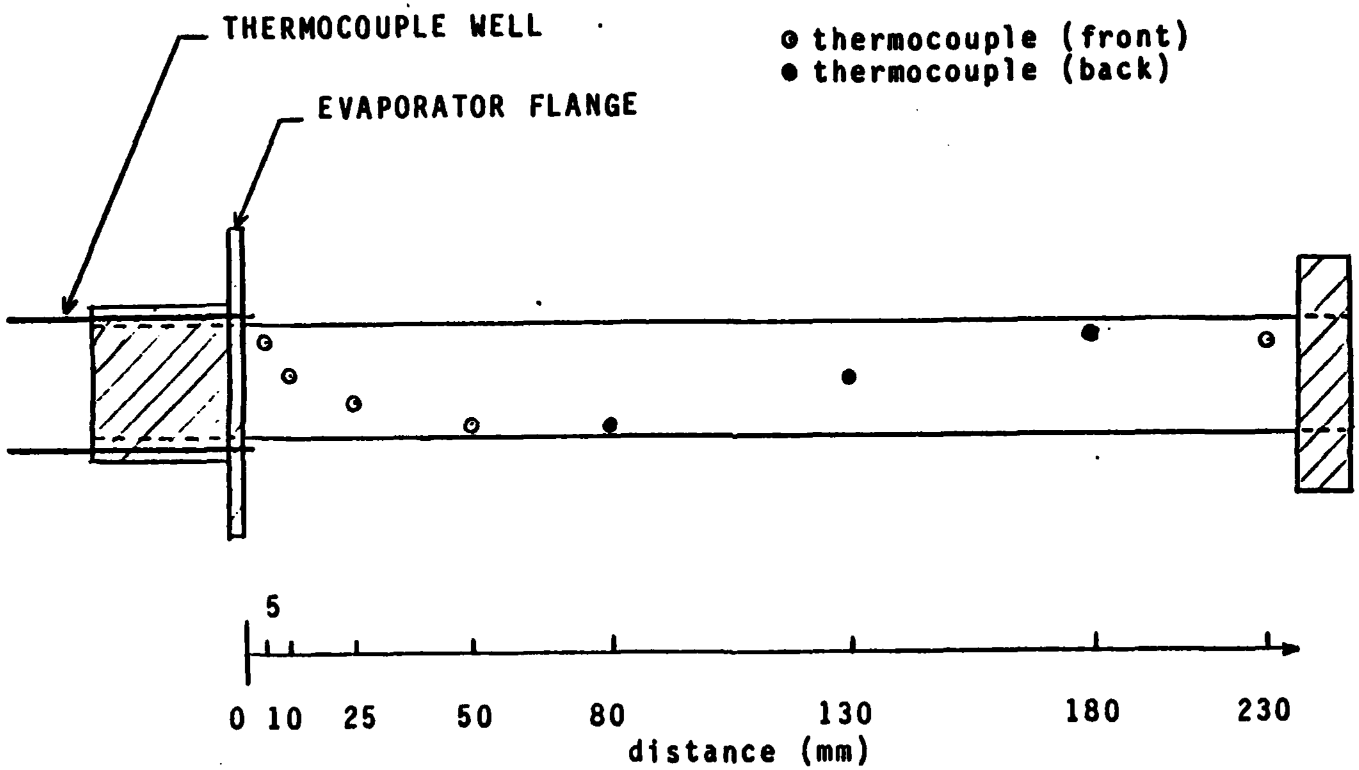

Figure 2.6 Thermocouple Locations on the Condenser

thermocouples was a major cause of delay in this thesis. An assembled condenser is shown in Figure 2.7.

The thermocouple signals were transmitted from the rotating system by a set of low noise, viscous slip rings containing mercury. The slip rings were refurbished and rewired for type-E application prior to use.

the thermocouple voltages (EMF) were monitored by a Hewlett Packard (HP) 3054a data aqusition system that was controlled by an HP9826 computer. An interactive program (Appendix C) was written to control the data aqusition and to anslyze the data during operation. Raw data and final results were stored on flexible disk for re-use at a later time. The computer aided data aqusition and analysis provided real time results and made it possible to detect problems or inconsistencies in the results as soon as they occurred.

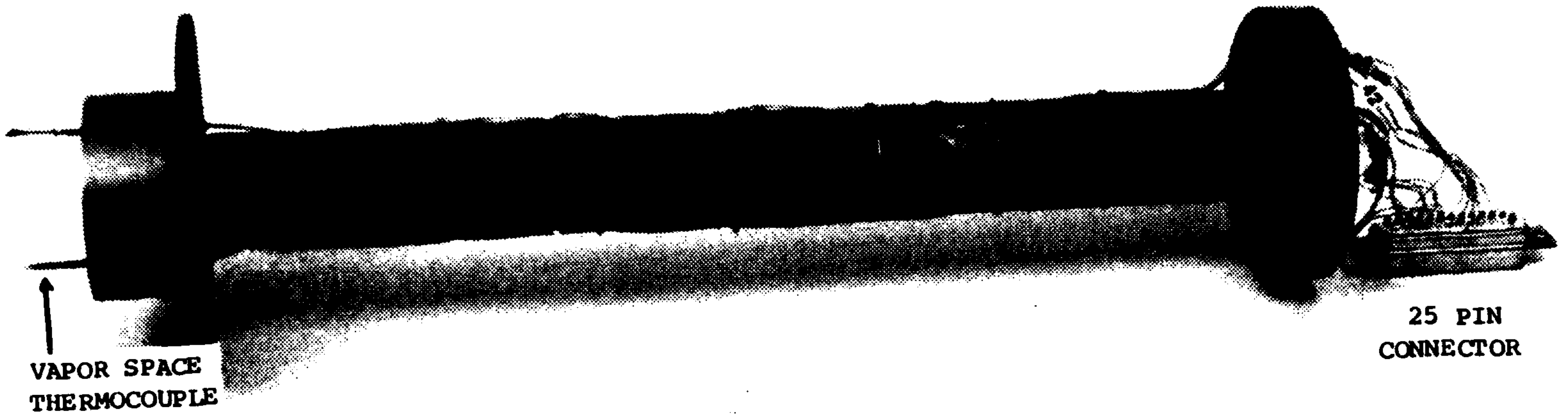

Figure 2.7 Photograph on an Assembled Condenser

III. EXPERIMENTAL PROCEDURES

The steps outlined in this section have been chosen to ensure consistent and repeatable results.

A. INSTALLATION AND PRETESTS

1. Inspect all O-rings and O-ring grooves to ensure they are clean and have no flaws.
2. Assemble the heat pipe system ensuring all joints are tight.
3. Install a test flange in place of the view windows; connect a pressure source to the flange and raise the pressure inside the heat pipe to 0.3 mP. Check all joints and surfaces for leaks using a soap solution. The surfaces should be thoroughly wetted and should be inspected several times over a five to ten minute period. Small leaks will appear in time as a mound of tiny foam bubbles. This simple test will easily detect leaks that will allow noncondensable gases to be inducted into the heat pipe which operates in a partial vacuum. Daniels and Williams [Ref. 5] have shown that noncondensable gases significantly reduce heat transfer rates in rotating heat pipes. Small leaks plagued this work from the beginning. leaks were found at several O-ring joints, in the thermocouple wells and in soldered and brazed connections on the heat pipe. Repeated attempts were therefore made to correct these dificulties, but these attempts were not always successful or lasting.
4. Disconnect the pressure source and install a vacuum test system. Evacuate the heat pipe; isolate the system and allow it to sit for at least 24 hours.

Monitor the vacuum in the system to ensure that there are no leaks. For the reasons stated above, it is imperative that the system be vacuum tight. Data taking was delayed for three weeks while leaks were being found and repaired. More leaks were found in the tubing and Swagelok fittings of the vacuum test system than in the rotating heat pipe.

B. PREPARATION OF THE HEAT PIPE INTERIOR

1. Remove the test flange and tilt the evaporator end down slightly to allow the cleaning fluids to run out of the evaporator. Place a pan under the evaporator to catch the cleaning fluid run off.
2. Using a stiff bristle brush, scrub the interior of the evaporator and the condenser with acetone. Rinse with distilled water by spraying and thoroughly wetting the interior of the evaporator and condenser.
3. Scrub the interior of the evaporator and the condenser with ethyl alcohol and rinse with distilled water.
4. Scrub the interior of the evaporator and the condenser with a hot solution (80 degrees Celsius) of equal parts ethyl alcohol and 50 percent aqueous sodium hydroxide. Rinse the interior thoroughly with distilled water. Observe the interior of the condenser to ensure that there is even wetting of the surface when water is sprayed on the surface.
6. The interior of the heat pipe system is now prepared for filmwise condensation.

C. FILLING

1. Tilt the condenser end down about 30 degrees.
2. Pour 300 ml of distilled water into the heat pipe.
3. Thoroughly dry the O-ring groove area. This will prevent condensation between the view windows.
4. Install the two view windows using two dry spacers rings between the inner and outer windows. A 1 mm wooden or plastic shim may used to center the windows.
5. The twelve view window retaining bolts are tightened sequentially. Proceeding clockwise, every fifth bolt is tightened to 20 inch pounds torque. Each bolt may be tightened several times during this process. Repeat the procedure, tightening each bolt to 30 inch pounds torque. The two step tightening procedure prevents cracking the view windows during the installation.

D. VENTING

This procedure is used to drive noncondensable gasses out of the working fluid and the heat pipe interior.

1. Energize the HP3054A data aquisition system and monitor one of the vapor space thermocouples.
2. Tilt the evaporator end down 30 degrees.
3. Remove the vent screw to prevent pressure build up during venting and to protect the view windows from cracking.
4. Set the power control to 13.5 and heat the system up to about 104 degrees C. Power settings above 14.0 during system warm up have cracked the interior view window.
5. When a steady plume of steam is observed at the vent, commence timing and allow the system to vent

for 10-15 minutes. Adjust the power control as necessary to maintain about 104 degrees C during the venting.

6. When venting is complete, install the vent screw and O-ring and immediately secure the power.
7. Tilt the heat pipe to the horizontal position.
8. Turn on the condenser cooling water to cool off the system. Violent boiling may be observed in the evaporator as the system starts to cool off.

E. RUNNING

1. Energize the HP9826 and the HP3054A. Load and start the data aquisition program.
2. Energize the tachometer and its digital display.
3. Open the valves and establish cooling water flow to the main bearing.
4. Open the needle valve on the oil dripper and adjust the oil flow to the main bearing for 2-3 drops per minute.
5. Turn on the cooling water and adjust the flow rate for a rotmeter reading of 50 percent. This is the same flow setting used in previous work and was chosen to reproduce the previous work as closley as possible.
6. Rotate the heat pipe by hand to ensure there is no binding.
7. Start the drive motor and raise the speed to 1100-1400 rpm and observe that a liquid annulus forms in the evaporator.
8. Adjust the speed to the desired rpm. In this work, as in all the previous work, data was taken at 700, 1400 and 2800 RPM.

9. At a given RPM, set the power control to the desired level. Select and monitor one of the vapor space thermocouple EMF's on the data aquistion voltmeter. Wait ten minutes for the system to reach steady state. Steady state conditions are achieved when the thermocouple EMF is nearly constant, fluctuating no more that +/- 4 microvolts.
10. Take five sets of data at each power setting. The results of the five data sets will be averaged to reduce the experimental uncertainties. A set of data consists of the RPM, the rotometer reading and an automatic sampling of all the thermocouple EMF's.
11. Repeat step 7-9. Data should be taken for both increasing and decreasing power settings to determine whether or not the heat pipe exhibits any hystersis.

F. DATA REDUCTION

The data aquistion program and sample outputs are found in appendix C. The data aquisition portion of the program performs the following steps for each data run:

1. Requests the entry of the RPM and the cooling water rotometer reading and stores these values for future use.
2. Samples each thermocouple EMF twenty times and stores the average reading for future use.
3. Displays the uncorrected temperature readings for all thermocouples in degrees Celsius. This makes it possible to monitor thermocouple status at each data run. The temperature distribution along the condenser wall can be seen.

The analysis portion of the program performs the following steps for each data run:

1. Corrects the cooling water inlet (Tci) and the cooling water outlet (Tco) temperatures using their respective calibration curves. The accuracy of the calibration is +/- 0.05 degree Celsius.
2. Computes Twall, the average of all condenser wall temperatures. Computes Ts, the average of the two vapor space temperatures. Computes Tavg, the average of the cooling water inlet and outlet temperatures.
3. Computes $\dot{m}$, the cooling water mass flow rate (Kg/sec) using the rotometer calibration curve and a density correction factor based on the cooling water inlet temperature. The accuracy of the mass flow rate calculation is +/- 0.005 Kg/sec.
4. Computes Q, the heat transfer rate to the cooling water by the following energy balance equation:

   $$Q = \dot{m}*Cp*(Tco-Tci) - Qf$$

   where Qf is the frictional heat generated in the system. Qf is determined for each test RPM by:

   $$Qf = \dot{m}*Cp*(Tco-Tci)$$

   at zero power. The Qf's are incorporated into the program and the approriate one is automatically selected based on the RPM entered at the beginning of each data set.
5. Displays the following output:

Q	watts
Ts-Tci	degree C
Ts- Twall	degree C
Twall-Tavg	degree C
Tco-Tci	degree C

6. Stores the outputs on disk for future use.

IV. PRESENTATION AND DISCUSSION OF RESULTS

A. GENERAL COMMENTS

Three different data runs were made for this thesis. Before the first data run was made, zero power data was taken at 700, 1400 and 2800 RPM and the frictional heat transfer rates (Qf) determined. The first data run was made on 7 May, 1983 using the smooth condenser. Data was taken for 10 power settings at 700 RPM, 7 power settings at 1400 RPM and 5 power settings at 2800 RPM. The second data run was made on 10 May, 1983 with the smooth condenser. Data was taken for 6 power settings at 700 RPM and 12 power setting at 1400 RPM. A 2800 RPM data run was attempted on 11 May, 1983, but all of the thermocouple wires pulled out of the connector and were damaged beyond repair. The third and last data run was made on 18 May, 1983 with the axial finned condenser. Data was taken for 16 power settings at 700 RPM. No data was taken beyond this point due to leakage in the heat pipe system.

The experimental results obtained are displayed graphically. For each condenser, at each test RPM, a plot of Heat Transfer Rate(Q) vs. the Temperature difference between the vapor space and the cooling water inlet (Ts-Tci) was made. The results of Weigel [Ref. 3] and Wagenseil [Ref. 4] are also plotted for comparison. On each figure, a typical uncertainty band is shown. The experimental results fluctuated more that the combined uncertainties of the measurement accuracies would predict. Observing the instrumentation during operation, the rotometer reading did not change. The temperatures in the cooling water fluctuated as much as 0.5 degrees when electrical power was supplied to the evaporator. This fluctuation was greater than the calibration accuracy and therfore was used for the uncertainty in the error analysis. It is suspected that there is incomplete mixing and unstable flow in the drain collecting box and the single thermocouple in the drain flow cannot provide an average bulk temperature. A second set of plots displays representative condenser wall temperature profiles for each condenser at each test RPM.

B. RESULTS OF THE SMOOTH WALL CONDENSER

Figure 4.1, 4.2 and 4.3 show the results of the smooth wall condenser at 700, 1400 and 2800 RPM. It was hoped that the smooth wall condenser data would correlate well with the work cf Weigel [Ref. 3] and Wagenseil [Ref. 4]. As seen in Figure 4.1, 4.2 and 4.3 the heat transfer rates obtained in this work were 10 to 80 percent lower than those obtained in previcus work. The heat transfer rates obtained in this work follow the the same trend observed in previous work. The increased heat transfer rates at the higer RPM's is the result of increased centrifugal acceleration flattening and thinning the condensate film on the inside of the condenser.

Experimental procedure was ruled out as a cause for the discrepancy between the results of this thesis and previous work. The same cleaning, filling and venting procedures were employed in each experiment. The presence of noncondensable cases is the most likely cause of the lower heat transfer rates obtained in this work. The noncondensable gases are drawn intc the heat pipe while it operates in a partial vacuum. Daniels and Williams [Ref. 5] have shcwn that the presence of noncondensable gases significantly reduces the heat transfer rates in a rotating heat pipe. A data run was attempted on 9 May, 1983, but was aborted because of erratic readings. An inspection of the system after the aborted run revealed a damaged O-ring in the

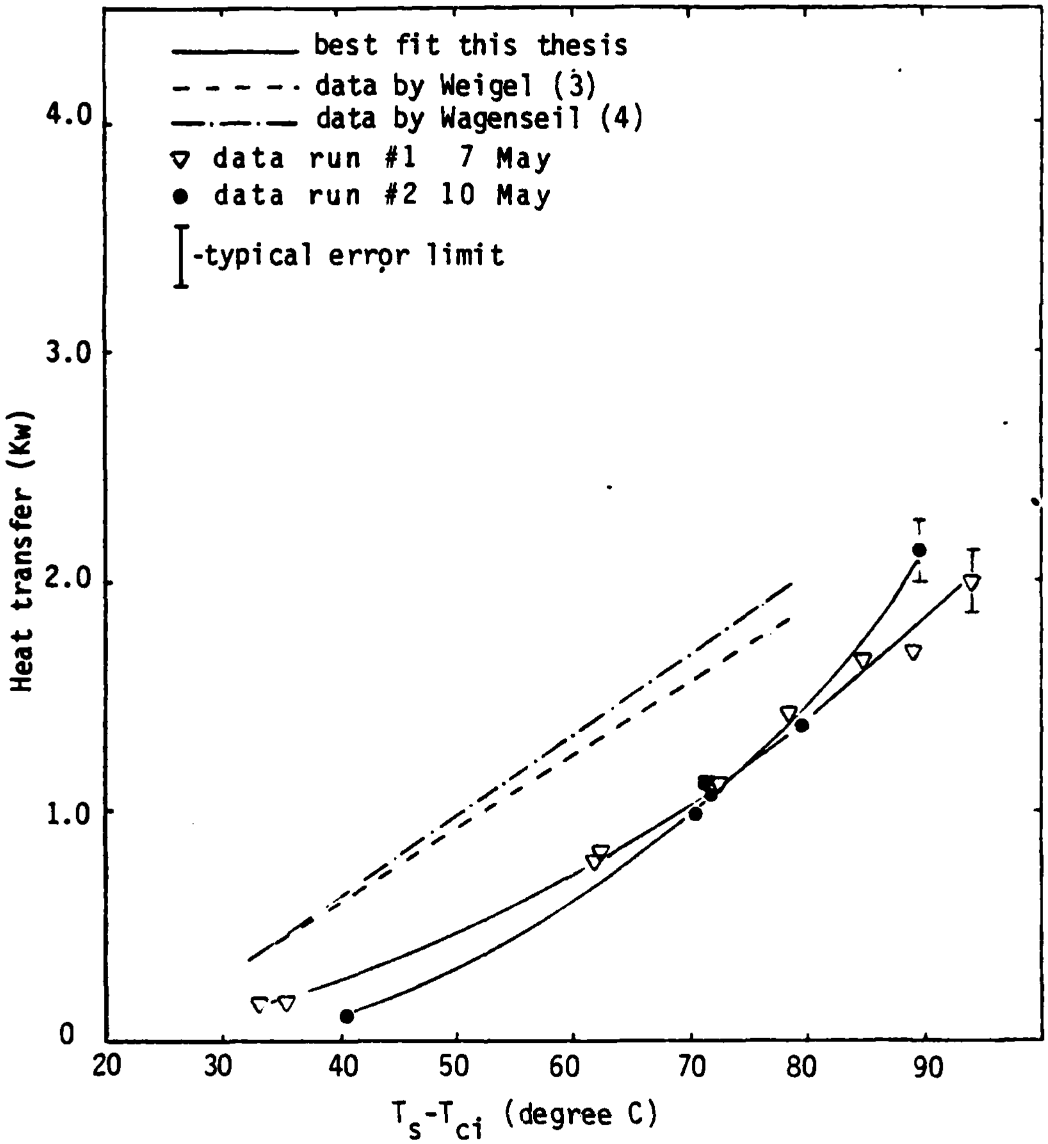

Figure 4.1 Smooth Condenser Thermal Performance at 700 RPM

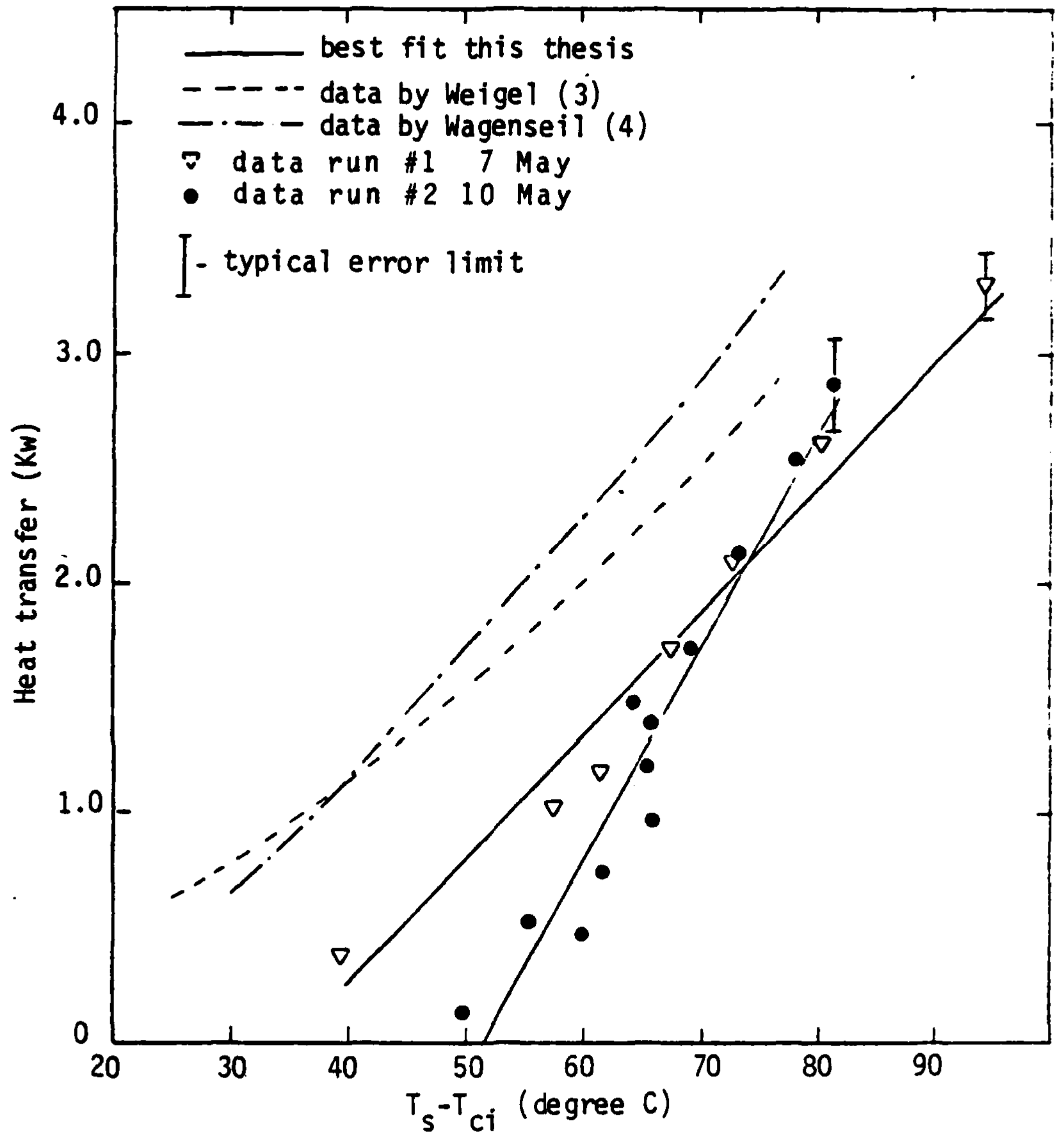

Figure 4.2 Smooth Condenser Thermal Performance at 1400 RPM

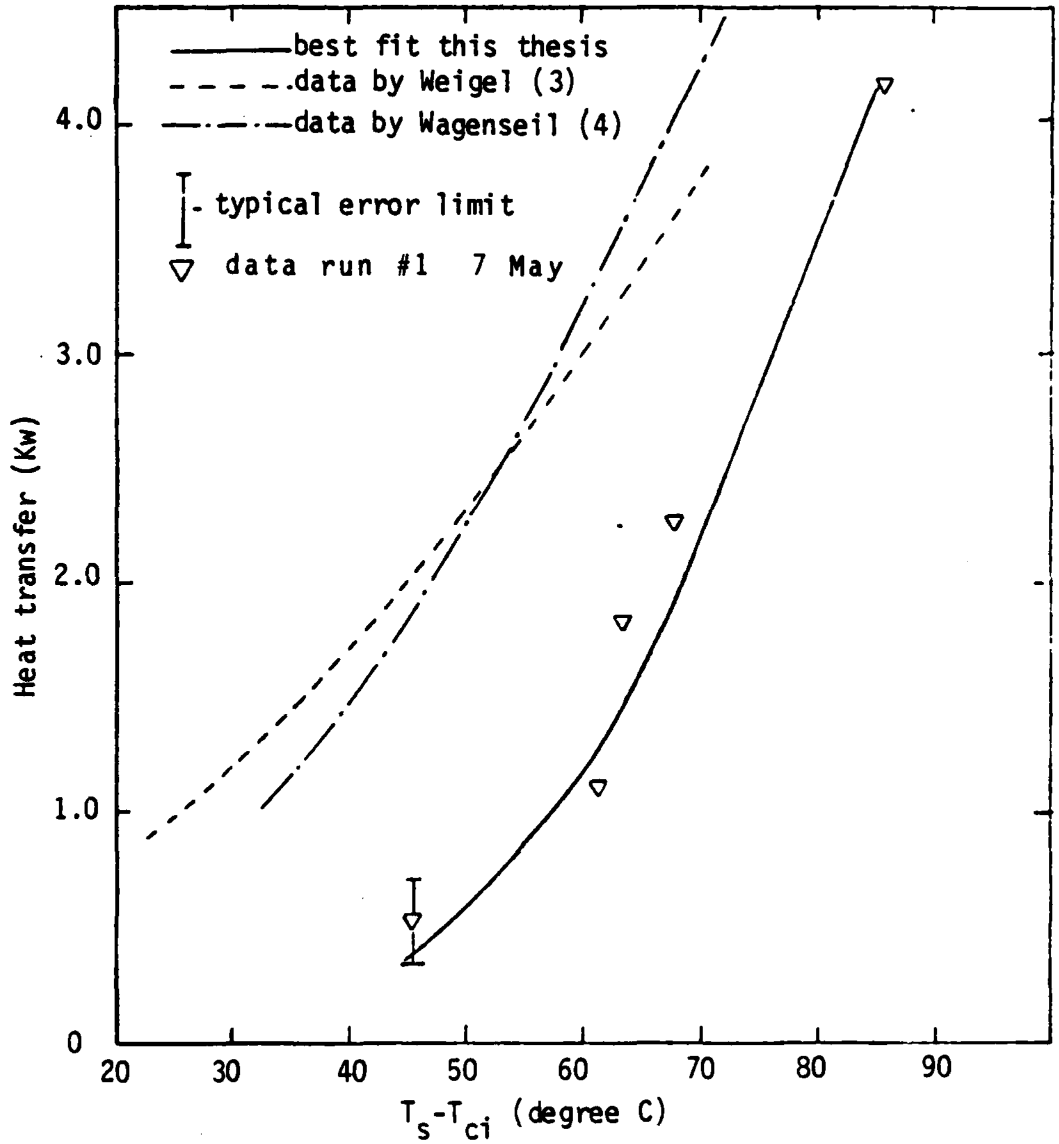

Figure 4.3 Smooth Condenser Thermal Performance at 2800 RPM

condenser vent. It is suspected that the O-ring may have been slightly damaged during the venting prior to data run #1 and further damaged during the venting prior to the aborted run. The O-ring had not been removed, but inspected in place prior to the aborted run. Following data run #2, a pressure test of the system was performed and a pin hole leak was found in a brazed patch in one of the flanges. This leak was not present when the system was pressure tested prior to run #2.

The condenser wall temperature profiles for the smooth condenser are shown in Figures 4.4, 4.5 and 4.6. The data is incomplete due to the failure of the thermocouples located at 130 and 230 mm from the evaporator. In spite of the missing data, some observations can be made.

At all RPM's and all power settings, the temperature closest to the evaporator appears to peak. It is suspected that this is caused by heat conducted through the main bearing flange from the evaporator and the frictional heat generated in the main bearing, as well as the heat removed from the vapor condensing on the inside of the tube. Ignoring the temperature at 5 mm, the remaining profiles are relatively flat out to 80 mm and then they drop off to a temperature very close to that of the cooling water. Without the temperature at 125 mm it is difficult to decide whether or not these results are similar to those observed by Daniels and Al-Baharnah [Ref. 6]. They have shown that a rotating heat pipe containing air and water has a flat temperature profile near the evaporator and the profile drops sharply in the region of noncondensable gas buildup and becomes flat again at a temperature close to that of the cooling water. The air is blanketing the condenser and preventing condensation in the region farthest from the evaporator. This phenomenon appears to be present at 1400 RPM in Figure 4.5 and appears to be more pronounced at 2800

RPM in Figure 4.6. This trend is reasonable since data was first taken at 700 RPM then at 1400 RPM and finally at 2800 RPM. Six to eight hours can elapse between the venting of the system and the completion of data taking. This is ample time to draw in air through even the smallest leak.

C. RESULTS OF THE AXIALLY FINNED CONDENSER

The heat transfer rates determined for the axially finned condenser are shown in Figure 4.7. The data points are connected in the sequence in which the data was taken. The erratic behavior of the data is attributed to a buildup of noncondensable gases during the four hour data run. A pressure test immediately after the data run revealed a leak at the O-ring, sealing the drive shaft. This leak was not present during the pressure test of the system prior to the data run. Figure 4.7 clearly shows the degrading effect that the buildup of non condensable gases has on heat transfer rates in the heat pipe. The trend with time shows that a greater and greater temperature difference between the vapor and the cooling water is required to transfer the same amount of heat. The condenser wall temperature profiles for data points A, B, C and D are shown in Figure 4.8.

Also shown in Figure 4.7 are the average heat transfer rates for the smooth condenser from this thesis and previous work [Ref. 3] and [Ref. 4]. Considering only the first seven data points in Figure 4.7 it appears that the axially finned condenser has heat transfer rates 30-40 percent greater than a smooth condenser of similar dimensions. It is suspected that centrifugal acceleration rapidly removes the condensate film from the sides of the fins and that the condensate depth between the fins is never great enough to fully cover the fin. Thus the fins with a thin condensate

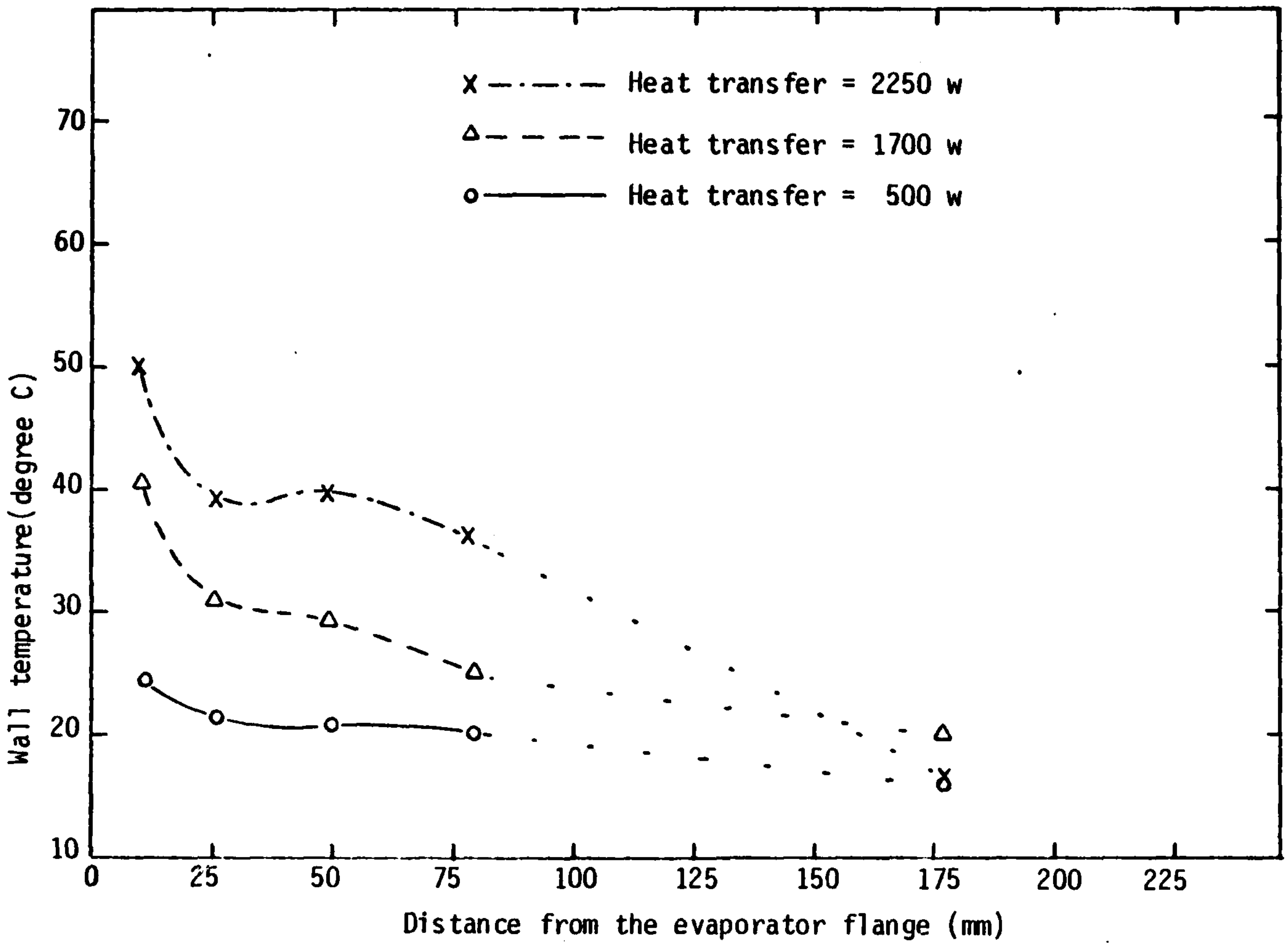

Figure 4.4 Smooth Condenser Temperature Profile at 700 RPM

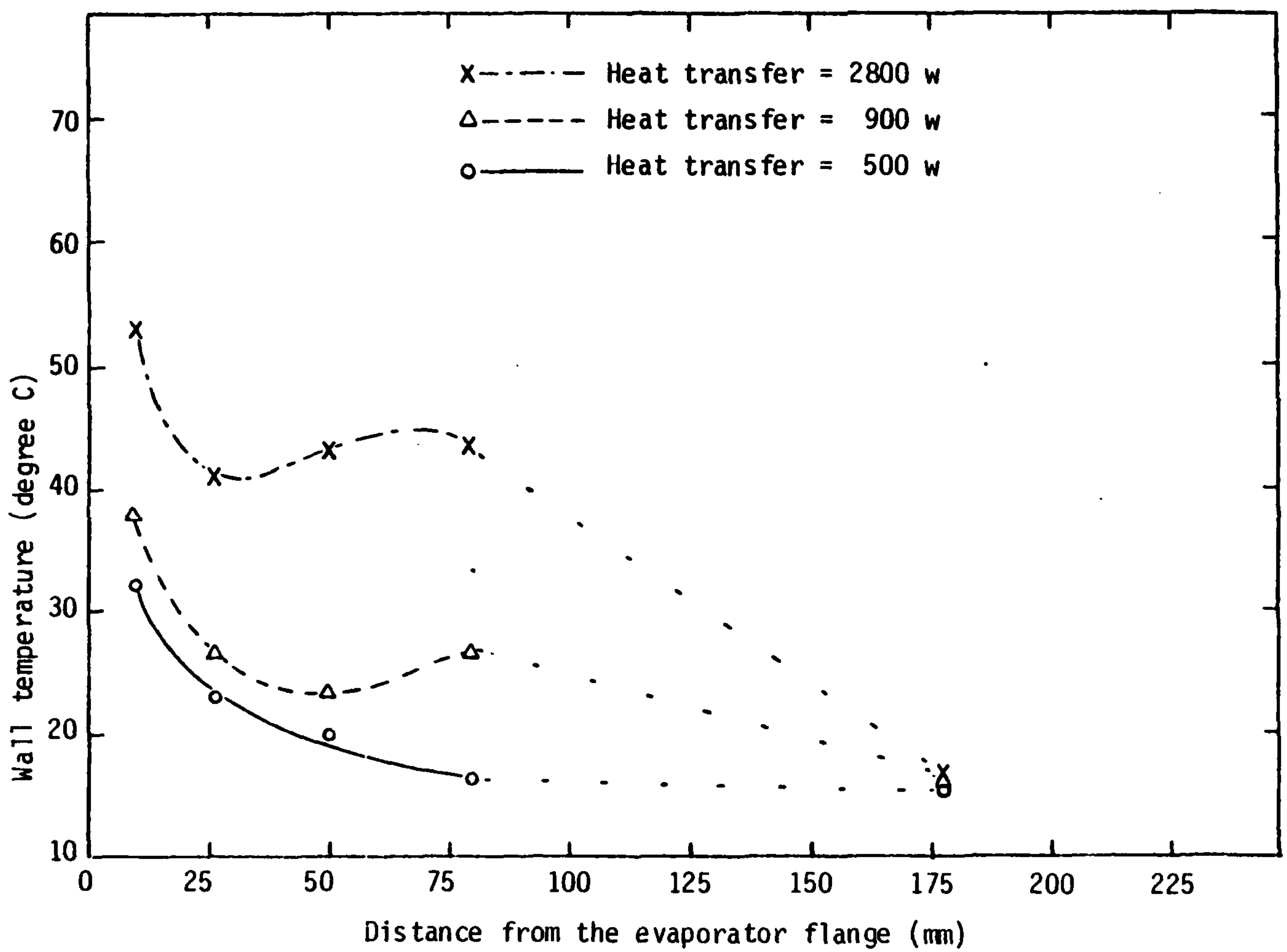

Figure 4.5 Smooth Condenser Temperature Profile at 1400 RPM

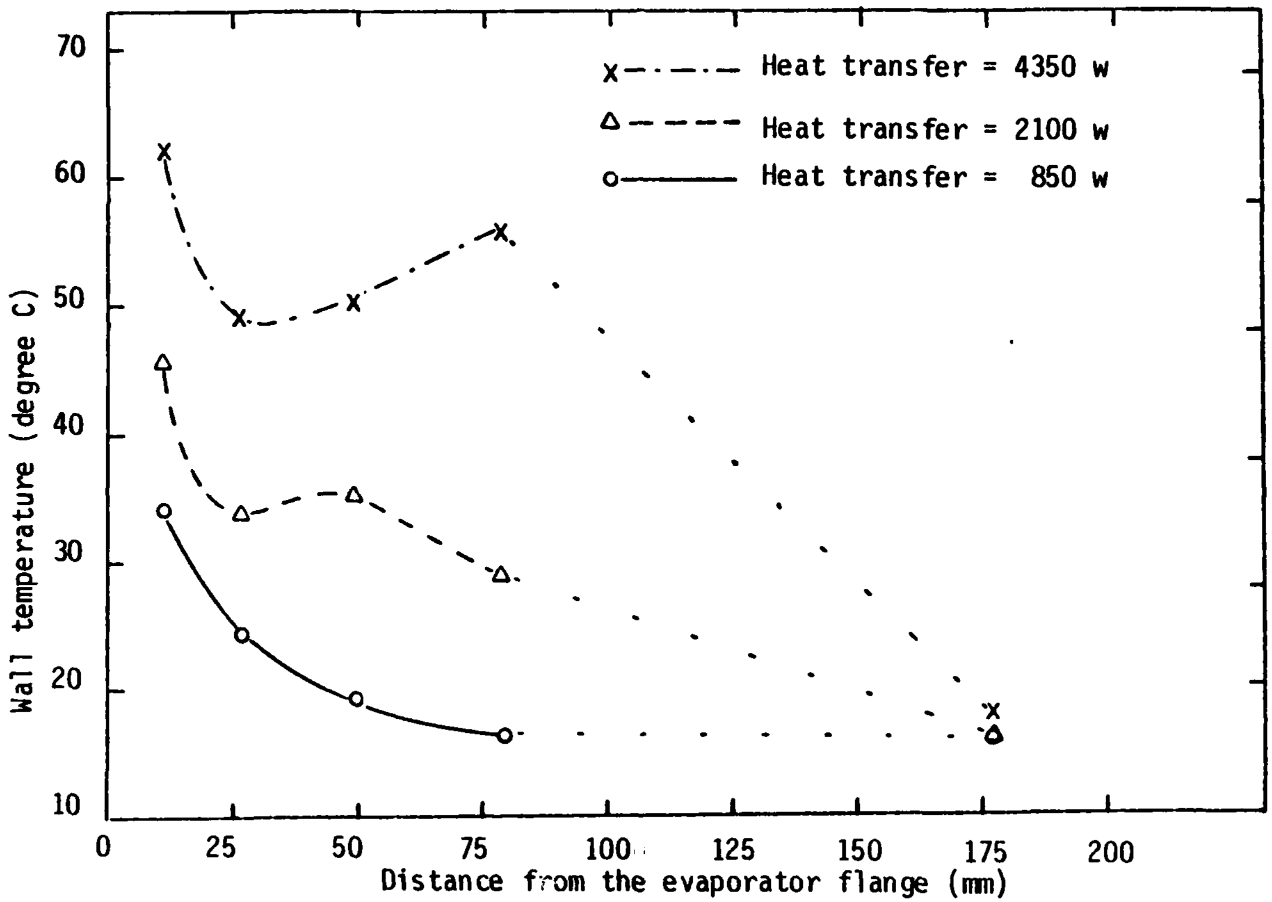

Figure 4.6 Smooth Condenser Temperature Profile at 2800 RPM

film will always be exposed to the vapor. This is in direct contrast to the inner surface of the smooth condenser which is always covered with a thicker layer of condensate. The thin film on the fins will have a higher heat transfer coefficient than the condensate layer that forms on the inside of the smooth condenser.

The four condenser wall temperature profiles shown in Figure 4.8 were selected to show how the buildup of noncondensable gases changes the temperature profile. The temperature closest to the evaporator is the highest, just as in the smooth condenser, and the same causes are suspected. The data for curve B was taken about 3 hours after the data for curve A. A sharp increase in the wall temperature near the evaporator is very apparent. It appears that the region of the heat pipe that is affected by noncondensable gases extends all the way to the evaporator since there is no flatness in any of the temperature profiles. The data for curve D was taken about 90 minutes after the data for curve C. The temperature profile for D is higher than C even though there is less heat transfer for curve D.

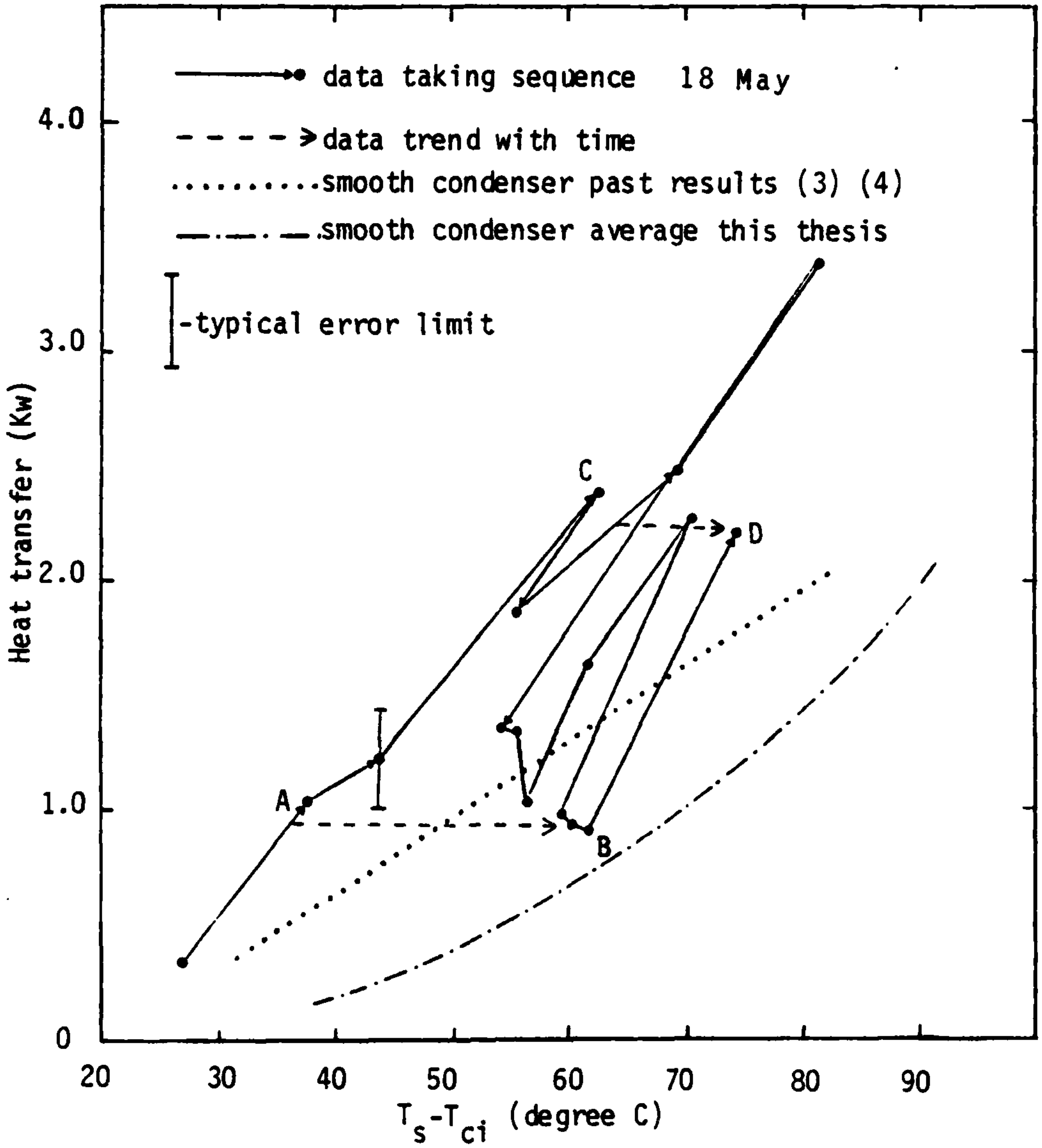

Figure 4.7 Finned Condenser Thermal Performance at 700 RPM

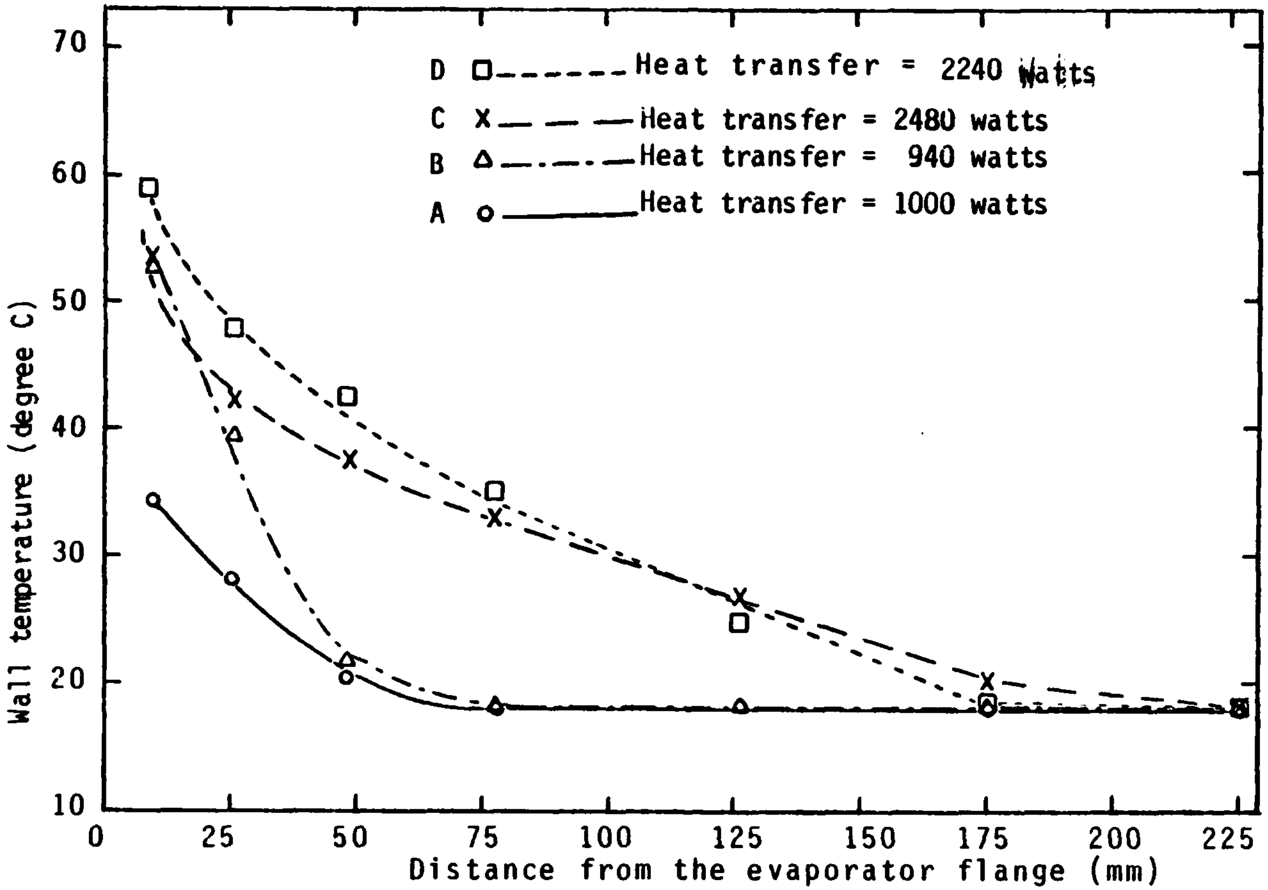

Figure 4.8 Finned Condenser Temperature Profile at 700 RPM

V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

The following conclusions have been made based on the experimental results obtained:

1. The use of straight axial fins in the condenser improves the heat transfer rate by 30-40 percent when compared to a smooth condenser of the same dimensions.

2. The smallest vacuum leak significantly reduces the heat transfer rate in a rotating heat pipe.

3. Total mixing does not occur in the drain collecting box. This is evident from the fluctuations observed in the cooling water outlet temperatures at all power settings.

4. The single thermocouples in the cooling water flow path do not provide an adequate sampling of the cooling water.

B. RECOMMENDATIONS

The following recommendations are made to improve the operational reliability of the rotating heat pipe system and the repeatability of results:

1. Replace the 0.12 mm diameter thermocouple wire with 0.254 mm diameter (30 gauge) wire. This will reduce the breakage of the wires during assembly, calibration and operation.

2. Replace the EIA 25 pin connectors with four Jones strip terminal boards with type-E terminal lugs. Design and build a new bracket to hold the terminal boards to the drive shaft. This arrangement will eliminate solder joints in the thermocouple circuit and should allow for more flexibility in the wiring during assembly.

3. Redesign the drive shaft flange to eliminate one of the O-ring joints. Figure 5.1 shows a possible design.

4. Design and build an all metal vacuum test system that has only soldered and O-ring joints. This will ensure that any leakage detected is in the heat pipe and not in the vacuum test system.

5. Replace the single cooling water thermocouples with five or more wired in parallel. Place the thermocouples to monitor various positions in the cooling water flow path. This should provide a good average temperature in the cooling water flow and reduce the fluctuations in the cooling water temperatures.

The following possible areas for future research should be considered:

1. Test the same condenser geometries used in this thesis again as well as additionall helical fin condensers. This will provide correlation with previous work and new data that should point towards an optimum condenser

2. Design and build a segmented cooling water system such as shown in Figure 5.2. Two condenser wall thermocouples should be located in each segment. A separate heat balance can be performed on the cooling water in each segment. The cooling water flow rate to each segment can be controlled and this can be used to control the condenser wall temperature distribution.

3. Design a new rotating heat pipe system to include the following:

   a. Segmented cooling system

   b. An induction heating system to eliminate the electric heater, brushes and collector rings

   c. Relocate the bearings so that frictional heat is not conducted to the condenser.

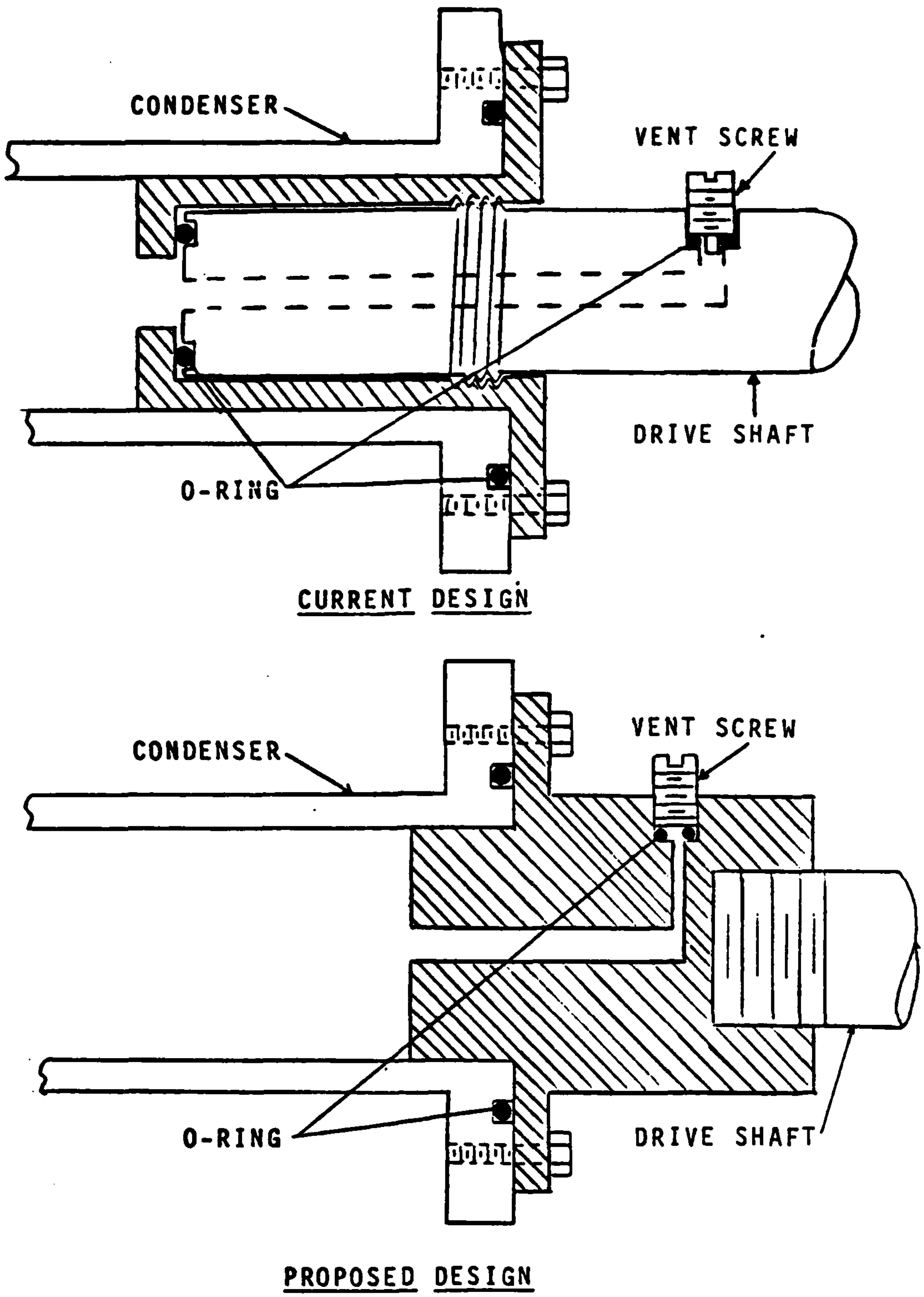

Figure 5.1 Proposed Change to Drive End Flange

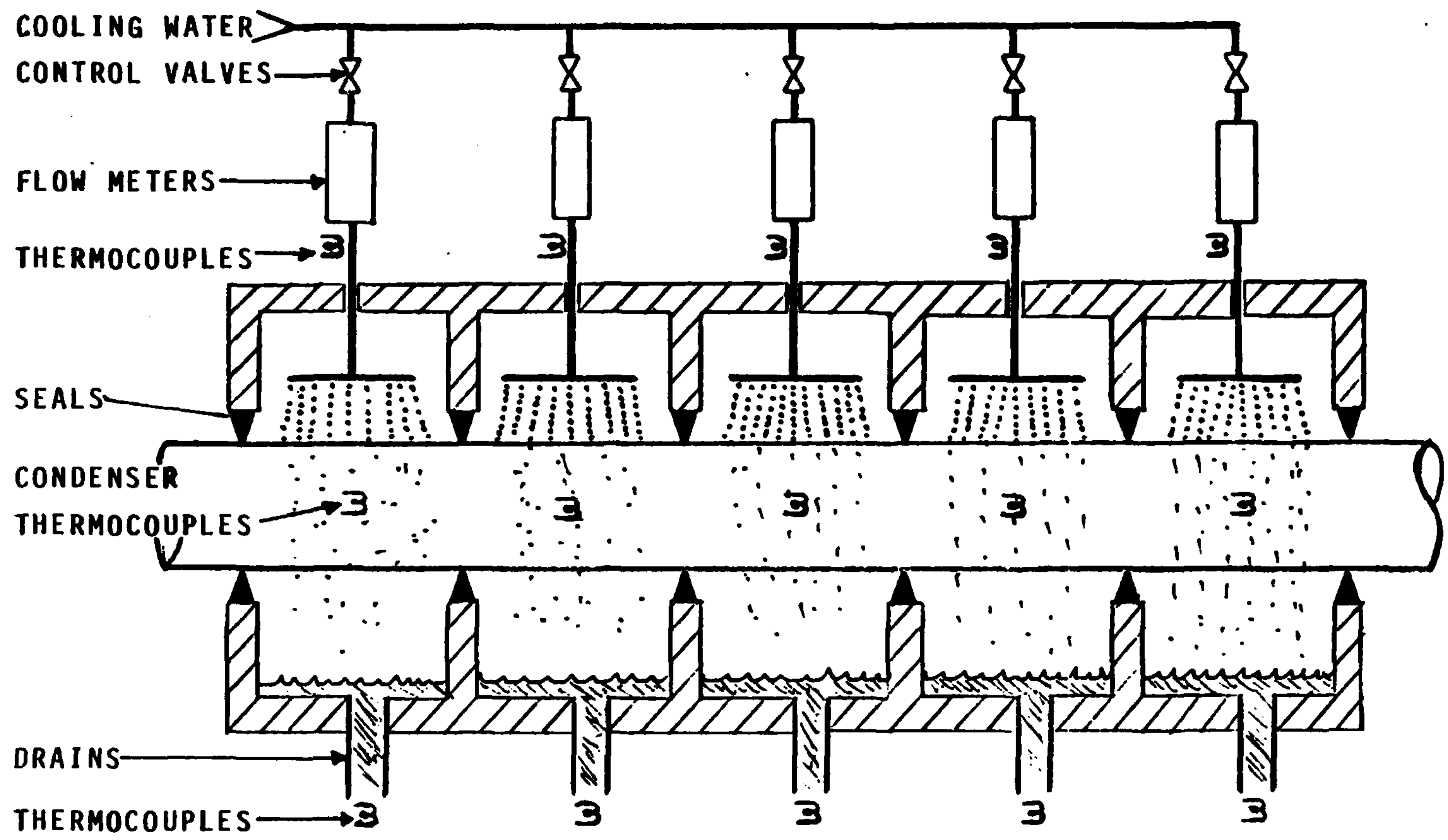

Figure 5.2 Proposed Segemented Cooling Water System

d. Separate the evaporator and condenser with an adiabatic section made from a low thermal conductivity material such as plexiglass. This section should most likely be conical in form.

APPENDIX A

UNCERTAINTY ANALYSIS AND SAMPLE CALCULATIONS

The uncertainty analysis of the experimental heat transfer rates was done by the method of Kline and McClintock [Ref. 7].

The following variables are the subject of the uncertainty analysis in this thesis:

Q = the heat transfer rate from the condensing vapor, watts

Qt = the total heat transfer rate to the cooling water when electrical power is supplied to the evaporator, watts

Qf = the frictional heat rate generated in the system when no electrical power is supplied to the evaporator, watts

Cp = the specific heat of water, kj/kg-C

$\dot{m}$ = the mass flow rate of the cooling water, kg/sec

Tci = the cooling water inlet temperature, degree C

Tco = the cooling water outlet temperature, degree C

ΔT = the cooling water temperature difference (Tco-Tci) when electrical power is supplied to the evaporator

ΔTf = the cooling water temperature difference (Tco-Tci) when no electrical power is supplied to the evaporator

The following equations are used for data analysis in this thesis:

$$\Delta T = Tco - Tci$$

$$\Delta Tf = Tco - Tci$$

$$Qt = \dot{m} Cp \Delta T$$

$$Qf = \dot{m} Cp \Delta Tf$$

$$Q = Qt - Qf$$

The uncertainties of the variables are designated: Wq, Wqt, Wqf, Wcp, Wm, Wti, Wto, Wt and Wtf respectively. The uncertainties are given by:

$$Wt = \left[\left(Wto\right)2 + \left(Wti\right)2\right]1/2$$

$$Wtf = \left[\left(Wto\right)2 + \left(Wti\right)2\right]1/2$$

$$\left(\frac{Wqt}{Qt}\right) = \left[\left(\frac{Wm}{\dot{m}}\right)^2 + \left(\frac{Wcp}{Cp}\right)^2 + \left(\frac{Wt}{T}\right)^2\right]1/2$$

$$\left(\frac{Wqf}{Qf}\right) = \left[\left(\frac{Wm}{\dot{m}}\right)^2 + \left(\frac{Wcp}{Cp}\right)^2 + \left(\frac{Wtf}{Tf}\right)^2\right]1/2$$

$$Wq = \left[\left(Wqt\right)2 + \left(Wqf\right)2\right]1/2$$

The following sample calculations are for the smooth condenser operating at 700 RPM. Data is shown, with its uncertainties, for zero power operation and operation with electrical power supplied to the evaporator.

	Zero Power	Power On
Cp (kj/kg-C)	4182	4182
Wcp	+/- 1	+/- 1
$\dot{m}$ (kg/sec)	0.2183	0.2183
Wm	+/- 0.005	+/- 0.005
Tci (C)	17.03	17.03
Wtci	+/- 0.05	+/- 0.5
Tco (C)	17.20	19.70
Wtco	+/- 0.05	+/- 0.5

The uncertainty for the zero power ΔTf is:

$$Wtf = \left[\left(0.05\right)2 + \left(0.05\right)\right]1/2$$

$$Wtf = \left[0.005\right]1/2$$

Wtf = +/- 0.07 degrees Celsius

The uncertainty for the power operation ΔT is:

$$Wt = \left[\left(0.5\right)2 + \left(0.5\right)2\right]1/2$$

$$Wt = \left[0.5\right]^{1/2}$$

$$Wt = +/- \ 0.7 \text{ degrees Celsius}$$

The frictional heat transfer rate, Qf, and its uncertainty are:

$$Qf = (0.2183)(4182)(17.20-17.03)$$

$$Qf = 155.198 \text{ watts}$$

$$Wqf = 155.198\left[\left(\frac{0.005}{0.2183}\right)^2 + \left(\frac{1}{4182}\right)^2 + \left(\frac{0.07}{0.17}\right)^2\right]^{1/2}$$

$$Wqf = 155.198\left[0.1753\right]^{1/2}$$

$$Wqf = +/- \ 64.65 \text{ watts}$$

The total heat transfer rate, Qt, and its uncertainty are:

$$Qt = (0.2183)(4182)(19.70-17.03)$$

$$Qt = 2437.525 \text{ watts}$$

$$Wqt = 2437.525\left[\left(\frac{0.005}{0.2183}\right)^2 + \left(\frac{1}{4182}\right)^2 + \left(\frac{0.7}{2.67}\right)^2\right]^{1/2}$$

$$Wqt = 2437.525\left[0.0005 + 0.000001 + 0.0021\right]^{1/2}$$

$$Wqt = 2437.525\left[0.095\right]$$

$$Wqt = +/- \ 231.6 \text{ watts}$$

The heat transfer rate from the vapor, Q, and its uncertainty are:

$$Q = 2437 - 155$$

$$Q = 2282 \text{ watts}$$

$$Wq = \left[\left(231\right)2 + \left(64\right)2\right]1/2$$

$$Wq = +/- 239.7 \text{ watts}$$

The fractional uncertainty in Q is:

$$\left(\frac{Wq}{Q}\right) = \frac{240}{2285}$$

$$\left(\frac{Wq}{Q}\right) = 0.105$$

APPENDIX B
CALIBRATION

In order to produce meaningful experimental results, the measurement of physical quantities used in the analysis must be as accurate as possible. The instruments used to measure the parameters in the cooling water energy balance were individualy calibrated.

A. ROTOMETER CALIBRATION

The rotometer was calibrated for volume flow rate in cubic meters per second by the following procedure. The cooling water flow was directed into a tank placed on a scale. The temperature of the cooling water was measured and the density of the water determined from subcooled liquid tables. The flow rate was adjusted to the desired rotometer reading and the time required to add 20 lbm to the tank was measured. Dividing the mass by the recorded time gave the mass flow rate. The mass flow rate was converted to volume flow rate by dividing by the density. A plot of volume flow rate vs. rotometer reading was made and a linear calibration curve derived by using a least squares fit of the data. The equations for volume flow rate as a function of rotometer reading and water density as a function of temperature were incorporated into the data aquisition and analysis program. The overall accuracy of the cooling water mass flow rate calculation in the program is +/- 0.5 percent.

B. THERMOCOUPLE CALIBRATION

All thermocouples were calibrated by immersing them into a Rosemont model 913A Calibration Bath and using a mercury in glass thermometer (accuracy +/-.028 degree C) as a standard.

1. Cooling water thermocouples (Tco &Tci)

The bath temperature was vairied up and down in 10 degree increments from 8 to 60 degree C. At each data point, the actual temperatures measured by the thermometer and the temperature measured by the data aquisition system were recorded and the difference between them computed. A plot of temperature difference vs. the data aquisition system's measured temperature was made and a linear calibration curve derived for each thermocouple by a least squares fit of the data. The calibration equations for Tco and Tci were incorporated into the data aquisition and analysis program. The accuracy of the calibration equations is +/- 0.05 degree C.

2. Condenser wall thermocouples

An attempt was made to calibrate the condenser wall and the vapor space thermocouples by immersing the assembled condensers into the calibration bath. Unfortunately there was not enough wire on the thermocouples to allow the 25 pin connectors to be located anywhere but directly over the bath. As the bath temperature was raised above 50 degree C, vapor escaping from the bath condensed on the connector causing erratic readings. Therefore, the condenser wall thermocouples were used only to provide qualitative information and could not be used for analytic calculations. The use of the Jones strip terminals rather than the 25 pin connectors will allow the terminal junctions to be placed well away from the vapor escaping from the bath and accurate calibrtion of the condenser wall and vapor space thermocouples will be possible in future work.

3. Vapor space thermocouples (Ts)

For the reasons stated, above the vapor space thermocouples could not be be calibrated. It was possible to check the accuracy of the vapor space thermocouples at 15 and 100 degree C. The thermocouple temperatures were found to agree with the bath temperature within +/- 0.5 degrees C. This accuracy was deemed sufficient for use in the graphical plot of heat transfer rate (Q) vs. thermal driving potential(Ts-Tci). In future work, where heat transfer coefficients are to computed, an accurate calibration of the vapor space thermocouples will be mandatory.

APPENDIX C

DATA AQUISITION AND ANALYSIS PROGRAM

```
TODAY'S DATE:  10 MAY 1983
TIME:  1400
PIPE CODE:  0

UNCORRECTED TEMPERATURES Deg.C
 40          82.222510502
 41          7.98597602346E+102
 42          17.3446620437
 43          7.98597602346E+102
 44          25.6402555781
 45          24.0017070503
 46          26.0840235919
 47          7.98597602012E+93
 48          44.9452493604
 49          82.7480678325
 50          16.9873841896
 51          18.2530745301

    Tci               Tco                 Tco-Tci
 17.1313542773      18.3761371673       1.24478289002

    Ts                Twall               Ts-Tci
 82.4852891673      27.6031795249       65.35393489

HEAT TRANSFER rate =  936.124982067  Watts


 1 DATA SETS ARE STORED IN FILE: 14SM1
```

Sample Program Output

```
10     !**********************************************************************
20     !                                                                     *
30     !          FILE NAME: HEAT_PIPE                                       *
40     !                                                                     *
50     !        DEVELOPED BY G.H. GARDNER                                    *
60     !                                                                     *
70     !                  12 MAY 83                                          *
80     ! THIS PROGRAM IS DESIGNED FOR USE WITH THE HP9826 COMPUTER AND THE   *
90     ! HP3054A DATA AQUISTION SYSTEM TO DO THE FOLLOWING:                  *
100    ! 1. GATHER DATA FROM THE ROTATING HEAT PIPE AND STORE IT IN A DATA FILE *
110    ! 2. REDUCE THE DATA, DISPLAY THE HEAT PIPE HEAT FLUX                 *
120    ! 3. STORE RESULTS ON DISK FOR LATER PLOTTING                         *
130    !                                                                     *
140    !**********************************************************************
150    !
160    ! THE FOLLOWING VARIABLES ARE USED IN THIS
170    ! PROGRAM:
180    !
190    !  Roto          ROTOMETER READING
200    !  Mf            COOLING WATER MASS FLOW (kg/SEC)
210    !  Rpm           RPM
220    !  Cp            AVERAGE SPECIFIC HEAT OF COOLING WATER (J/kg-DEG K)
230    !  Emf(I)        AN ARRAY CONTAINING THE THERMOCOUPLE VOLTAGES
240    !  Tci           COOLING WATER INLET TEMP (DEG.C)
250    !  Tco           COLLING WATER OUTLET TEMP (DEG.C)
260    !  Ts            VAPOR SPACE TEMP (DEG.C)
270    !  Twall         AVERAGE HEAT PIPE OUT SIDE WALLL TEMP (DEG.C)
280    !  Tavg          AVERAGE COOLING WATER TEMP (DEG.C)
290    !  Del_t         Ts-Tci, USED IN ONE OF THE PLOTS
300    !  Del_t2        Ts-Twall
```

```
310   !  Del_t3      Twall-Tavg
320   !  Q           HEAT TRANSFER RATE (W)
330   !  Qf          HEAT GENERATED BY BEARING FRICTION (W)
340   !  T(I)        TEMPS MEASURED ON HEAT PIPE WALLS, STEAM SPACE, COOLING H20
350   !  Aa(0)        INNER SURFACE AREA OF SMOOTH PIPE (m^2)
360   !  Aa(1)       INNER SURFACE AREA OF 16 SPIRAL FIN PIPE (m^2)
370   !  Aa(2)       INNER SURFACE AREA OF 14 SPIRAL FIN PIPE  (m^2)
380   !  Aa(3)       INNER SURFACE AREA OF 32 SPIRAL FIN PIPE (m^2)
390   !  Aa(4)       INNER SURFACE AREA OF 22 STRAIGHT FIN PIPE (m^2)
400   !  Ar          AREA RATIO: AN/A0
410   !  Uo          OVERALL HEAT TRANSFER COEFFICIENT (WATT/SQ.m-K) OF SMOOTH
420   !              PIPE
430   !  Ua          OVERALL HEAT TRANSFER COEFFICIENT (WATT/SQ.m-K) OF FINNED
440   !              PIPE
450   !  Roh         DENSITY OF COOLING WATER (KG/M^3)
460   !
470   !
480   !     ****************************
490   !  PUT DIMENSION, COMMON AND DATA STATEMENTS
500   !  IN THE PROGRAM AT THIS POINT
510   !     ****************************
520   DIM Emf(11),T(11),A(9),Aa(4)
530   DATA .104967248,17189.45282,-282639.085,12695339.5,-448703084.6,1.10866E10
540   DATA -1.76807E11,1.1842E12,-9.19278E12,2.06132E13
550   READ A(*)  !** reads in coefficients for type-E thermocouple equation
560   DATA 10,11,12,13,14
570   READ Aa(*)
580   BEEP
590   INPUT "ENTER TODAYS DATE",Date$
600   PRINT "TODAY'S DATE:  ";Date$
```

```
610   BEEP
620   INPUT "ENTER TIME",Time$
630   PRINT "TIME:  ";Time$
640   BEEP
650   INPUT "ENTER PIPE CODE",Code
660   PRINT "PIPE CODE: ";Code
670   Area=Aa(Code)
680   !
690   !  ****************************
700   ! USE THIS SPACE FOR INITIALIZING
710   ! ANY VARIABLE THAT ARE NECESSARY
720   !  ****************************
730   !
740   J=0      ! this is a counter used if data is being retrieved from disk file
750   Jj=0     ! this is a counter used to control the printing of the uncorrecte
d
760            ! temperatures
770   BEEP
780   INPUT "ENTER INPUT MODE (1=3054A,2=FILE)",Im
790   IF Im=1 THEN
800   BEEP
810   INPUT "GIVE A NAME FOR THE NEW DATA FILE",D_file$
820   CREATE BDAT D_file$,20
830   ELSE
840   BEEP
850   INPUT "GIVE THE NAME OF OLD DATA FILE",D_file$
860   BEEP
870   INPUT "ENTER # OF DATA RUNS STORED",Nrun
880   END IF
890   ASSIGN @File TO D_file$
```

```
900   !
910   !       *****************************
920   !      DATA GATHERING COMMANDS
930   !       *****************************
940   !
950   ! This section will take twenty readings foreach thermocouple and average
960   ! them.  This is done to reduce data scattering.  The average thermocouple
970   ! voltage is then stored on disk in the file name given above so that it
980   ! can be used again.
990   INPUT "GIVE A NAME FOR FILE TO HOLD PLOTTING DATA:",Plot_d$
1000  CREATE BDAT Plot_d$,10
1010  ASSIGN @Filep TO Plot_d$
1020  IF Im=1 THEN
1030  BEEP
1040  INPUT "ENTER RPM",Rpm
1050  PRINT " *******"
1060  PRINT "DATA SET # ",Jj+1
1070  PRINT
1080  PRINT "RPM: ";Rpm
1090  BEEP
1100  INPUT "ENTER ROTOMETER READING",Roto
1110  PRINT "ROTOMETER READING: ";Roto
1120  OUTPUT 709;"AR AF40 AL51 VR1"
1130  FOR I=0 TO 11
1140  Emf(I)=0
1150  OUTPUT 709;"AS SA"
1160  E=0
1170  FOR L=0 TO 19
1180  ENTER 709;E
1190  Emf(I)=Emf(I)+ABS(E)
```

```
1200 NEXT L
1210 Emf(I)=Emf(I)/20
1220 NEXT I
1230 OUTPUT @File;Rpm,Roto,Emf(*)
1240 ELSE
1250 BEEP
1260 ENTER @File:Rpm,Roto,Emf(*)
1270 END IF
1280 J=J+1
1290 Jj=Jj+1
1300 !
1310 !*************************
1320 !This section determines the bearing friction correction factor to be used
1330 !in the analysis
1340 !*************************
1350 !
1360 IF Rpm=700 THEN Qf=158
1370 IF Rpm=1400 THEN Qf=204
1380 IF Rpm=2800 THEN Qf=282
1390 !
1400 !  ********************
1410 ! This section converts the thermocouple voltages to temperature Deg.C
1420 !
1430 FOR I=0 TO 11
1440 T(I)=0
1450 Emf(I)=ABS(Emf(I))
1460 FOR K=0 TO 9
1470  T(I)=T(I)+A(K)*Emf(I)^K
1480 NEXT K
1490 NEXT I
```

```
1500  IF Jj=1 THEN
1510  PRINT "UNCORRECTED TEMPERATURES Deg.C"
1520  FOR M=0 TO 11  ! This loop prints the uncorrected temperatures.
1530  PRINT M+40,T(M)
1540  NEXT M
1550  END IF
1560  !
1570  Ts=(T(0)+T(9))/2            ! T IN VAPOR SPACE
1580  Tsum=0
1590  Nn=0
1600  FOR I=1 TO 8
1610   IF T(I)<100 THEN
1620   IF T(I)>10 THEN
1630   Nn=Nn+1
1640   Tsum=Tsum+T(I)
1650   END IF
1660   END IF
1670   NEXT I
1680  Twall=Tsum/Nn
1690  Tci=-.2148294+1.03486638*T(10)-.00080912124*T(10)^2
1700  Tco=.118132221+1.002347201*T(11)-.0001137938*T(11)^2
1710  PRINT
1720  PRINT "     Tci                 Tco                        Tco-Tci"
1730  PRINT Tci,Tco,Tco-Tci
1740  PRINT
1750  PRINT "     Ts                  Twall                      Ts-Tci"
1760  PRINT Ts,Twall,Ts-Tci
1770  PRINT
1780  !
1790  !*******************************************
```

```
1800  !  THIS IS THE ANALYSIS PORTION OF THE PROGRAM
1810  !*****************************************
1820  !
1830  ! CALCULATE THE DENSITY OF THE COOLING WATER
1840  Roh=1000.073818+.0273614*Tci-.006429147*Tci^2+.00002153167*Tci^3
1850  !
1860  ! CALCULATE THE MASS FLOW RATE: Mf
1870  !
1880  Mf=Roh*(6.3948461E-6+4.2553734E-6*Roto)
1890  !
1900  ! NOW FIND THE AVERAGE COOLING BOX TEMPERATURE
1910  !
1920  Tavg=(Tci+Tco)/2
1930  !
1940  ! NOW FIND THE AVERAGE SPECIFIC HEAT OF WATER
1950  !
1960  Cp=4221.790953-3.442282*Tavg+.08713516*Tavg^2-.0006781436*Tavg^3
1970  !
1980  ! NOW COMPUTE THE HEAT FLUX FROM THE PIPE
1990  !
2000  Q=Mf*Cp*(Tco-Tci)-Qf
2010  PRINT "HEAT TRANSFER rate = ";Q;" Watts"
2020  PRINT
2030  PRINT
2040  !
2050  !NOW FIND THE DELTA-T USED FOR PLOTS
2060  !
2070  Del_t=Ts-Tci
2080  Del_t2=Ts-Twall
2090  Del_t3=Twall-Tavg
```

```
2100 Del_t4=Tco-Tci
2110 !
2120 !NOW STORE THE PLOTTING INFO ON DISK
2130 !
2140 OUTPUT @Filep;Q,Del_t,Del_t2,Del_t3,Del_t4
2150 IF Im=1 THEN
2160 INPUT "WILL THERE BE ANOTHER DATA RUN? (1=YES, 0=NO)",Go_on
2170   IF Go_on=1 THEN 1020
2180 ELSE
2190 IF Jj=5 THEN Jj=0
2200 IF J<Nrun THEN 1020
2210 END IF
2220 ASSIGN @File TO *
2230 ASSIGN @Filep TO *
2240 PRINT
2250 PRINT
2260 IF Go_on=0 THEN PRINT J;"DATA SETS ARE STORED IN FILE: ";D_file$
2270 END
```

LIST OF REFERENCES

1. Gray, V. H., "The Rotating Heat Pipe---A Wickless Hollow Shaft for Transferring High Heat Fluxes", ASME Paper No. 69-HT-19, 1969

2. Daly, T.J., The Experimental Design and Operation of a Wickless Heat Pipe, M. S. Thesis, Naval Postgraduate School, Monterey, California, June 1970

3. Weigel, H. J., Heat Transfer Performance of a Rotating Heat Pipe Using Different Condensers and Working Fluids, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1979

4. Wagenseil, L. L., Heat Transfer Performance of Rotating Heat Pipes, M. S. Thesis, Naval Postgraduate School, Monterey, California, September 1976

5. Daniels, T. C. and Williams, R. J., "Theoretical and Experimental Analysis of Non-Condensable Gas Effects in a Rotating Heat Pipe", Proceedings of the 2nd International Heat Pipe Conference, Bologna Italy, 1976

6. Daniels, T. C. and Al-Baharnah, N. S. "Temperature and Heat Load Distribution in Rotating Heat Pipes" AIAA Journal, Vol. 18, pp. 202-207, 1980

7. Kline, S. J. and McClintock, F. A., "Describing Uncertainties in Single Samlple Experiments", Mechanical Engineering p. 3, January 1953

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